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CONCEPTUAL DESIGN

RANKINE-CYCLE POWER SYSTEM WITH ORGANIC WORKING FLUID AND RECIPROCATING ENGINE FOR PASSENGER VEHICLES

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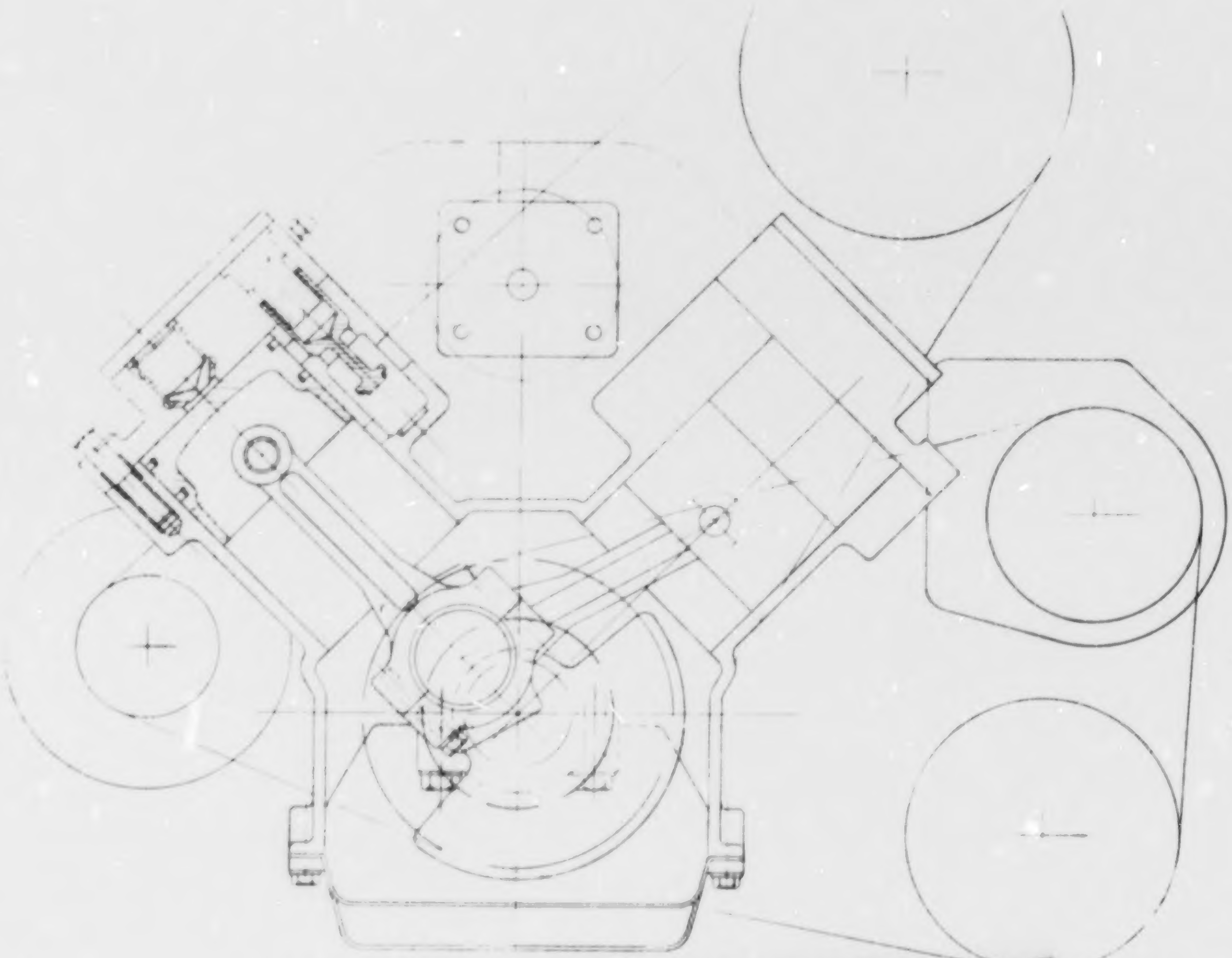

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A conceptual design has been prepared of a Rankine-cycle power system with organic working fluid and reciprocating engine for a low-emission automotive propulsion powerplant. The goal of the study was development of a system competitive in cost, performance, and driver convenience with the internal combustion engine system using current technology wherever possible. The component designs and characteristics are presented. The complete 100 net shaft horsepower system is packaged in the engine compartment of a 1969 Ford Fairlane; the predicted performance characteristics are presented. The system is closely competitive, in 0 - 60 mph acceleration time and in level-grade top speed, with a 302 cubic inch displacement internal combustion engine with three-speed transmission. The fuel economy in customer-average mpg is approximately 20% less than the 302 cubic inch internal combustion engine. The use of thiophene as a working fluid, with a moderate maximum cycle temperature of 550 degrees F, permits a significant cost reduction relative to the equivalent steam system. This reduction may permit the Rankine-cycle to be competitive with the equivalent internal combustion system, particularly since the stricter emission level standards will require significant cost increases in the future in the internal combustion system. Projection of current burner data indicates a strong potential for emission levels significantly less than the projected 1980 federal standards for all three of the major pollutants; unburned hydrocarbons, carbon monoxide, and nitric oxide.

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Thiophenes
Gas Expanders
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RANKINE-CYCLE POWER SYSTEM WITH ORGANIC WORKING FLUID AND RECIPROCATING ENGINE FOR PASSENGER VEHICLES

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1. ABSTRACT

A conceptual design has been prepared of a Rankine-cycle power system with organic working fluid and reciprocating engine for a low-emission automotive propulsion powerplant. The goal of the study was development of a system competitive in cost, performance, and driver convenience with the internal combustion engine system using current technology wherever possible.

The component designs and characteristics are presented. The complete 100 net shaft horsepower system is packaged in the engine compartment of a 1969 Ford Fairlane; the predicted performance characteristics are presented. The system is closely competitive, in 0 - 60 mph acceleration time and in level-grade top speed, with a 302 cubic inch displacement internal combustion engine with three-speed transmission. The fuel economy in customer-average mpg is approximately 20% less than the 302 cubic inch internal combustion engine.

The use of thiophene as a working fluid, with a moderate maximum cycle temperature of 550°F, permits a significant cost reduction relative to the equivalent steam system. This reduction may permit the Rankine-cycle system to be competitive costwise with the equivalent internal combustion system, particularly since the stricter emission level standards will require significant cost increases in the future in the internal combustion system.

Projection of current burner data indicates a strong potential for emission levels significantly less than the projected 1980 federal standards for all three of the major pollutants: unburned hydrocarbons, carbon monoxide, and nitric oxide.

2. SUMMARY

CONCEPTUAL DESIGN OF A RANKINE-CYCLE PROPULSION SYSTEM FOR PASSENGER VEHICLES

2.1 INTRODUCTION

Great emphasis is being placed on air pollution reduction in the United States. Since the internal combustion engine powered automobile represents the largest contributor to air pollution, reduction of emissions from this source represents an extremely important requirement for improving our air quality; strong federal pressure is being, and will continue to be, exerted on the automobile industry to reduce pollutant emissions.

The system which, comparatively, offers the greatest potential for absolute minimum emission of particulates, unburned hydrocarbons, NO, and CO in a system with range and power equivalent to the internal combustion engine is the Rankine-cycle system. Because of the strong potential of the Rankine-cycle system as the automotive propulsion system with the lowest possible emission level, the Motor Vehicle Research Division, National Air Pollution Control Administration, Public Health Service, Department of Health, Education, and Welfare is supporting the development of such a system. A detailed conceptual design of the system has been completed and the hardware development of the more critical components has been started. The emphasis in the development is on acceptable performance, packaging, and overall system cost when compared to the equivalent internal combustion system.

In this summary, a description is presented of the complete design and characteristics of a Rankine-cycle automotive propulsion system with thiophene working fluid and reciprocating engine, installed in a 1969 Ford

Fairlane. Projections of emission levels are also made, using results of a proprietary burner developed at Thermo Electron Corporation.

2.2 COMPONENT DESCRIPTIONS

2.2.1 System Design Point Characteristics

The system component sizes are based on the peak power requirements for an intermediate-sized automobile (Ford Fairlane) to give 0 - 60 mph acceleration of 15 seconds or better and top speed of ~100 mph. Using these criteria, the engine shaft horsepower less feedpump power was taken to be 103.2 hp, obtained from a 184 CID vapor engine. In Table 1, the design point characteristics of the propulsion system are presented. Figure 2.1 shows a simplified flow schematic and the cycle conditions for the thiophene working fluid on a T-S diagram. It will be noted that thiophene, due to its almost vertical saturated vapor line on the T-S plot and the small amount of superheat (~40°F) in the boiler outlet vapor, requires a relatively small regenerator compared to other organic working fluids; the ratio of boiler heat transfer rate to regenerative heat transfer rate is 0.16 for the cycle conditions.

2.2.2 Engine (Expander) Design

Establishing the dimensions of a new engine design depends, among other things, on the prediction of the indicated and mechanical efficiencies of the engines at the design condition. Consequently, a detailed analysis was carried out to determine these efficiencies as functions of piston speed and load, using measured efficiencies obtained with the 5 hp engine on test at Thermo Electron as a check. As a result of the analysis, it is possible to plot engine efficiency versus piston speed at various loads. One such plot is shown in

Figure 2.2 for an indicated mean effective pressure (IMEP) of 125 psi. The rapid drop in efficiency with piston speeds above 1000 ft/min occurs due to inlet valve losses.

From this analysis, a piston speed of 1000 ft/min was selected for the design condition of 103 bhp at a vehicle speed of 95 mph. The reduction in engine size which could be realized by selecting a higher piston speed would probably be more than lost in boiler and condenser size increases due to lower overall cycle efficiency.

The IMEP and BMEP are determined by the cycle design condition, and the BMEP and the piston speed determine the piston area required to develop the desired horsepower. A 90° V of four cylinders was selected as being reasonably compact without either an excessive number of moving parts or excessive torque variation. The V design results in a short engine for a given number of cylinders and facilitates packaging of the system. With four cylinders, the resulting bore is 4.42 inches. The mean piston speed (1000 ft/min at design) and the engine speed are related by the expression

$$S = 2\ LN$$

where S = mean piston speed, L = stroke, and N = rpm. The selected design point rpm of 2000, based on a reasonable bore-to-stroke ratio (1.47) and on valve train dynamics, results in a stroke of 3.0 inches. The basic engine dimensions and specifications are given in Table 2.1, and cross-sectional views of the engine are presented in Figures 2.3 and 2.4 with hydraulic engine valving, slipping-clutch transmission, and feedpump incorporated.

All materials are identical to those now used in automotive internal combustion engines. The Rankine-cycle expander differs from the current automotive internal combustion engine in two very important aspects of its design: the inlet valving and the bearing design.

2.2.2.1 Variable Cut-Off Inlet Valving. The importance of having a valving system with variable cut-off is established in a later part of this paper. Apart from the difficulty in varying the cut-off, the valving problem is considerably more severe than in internal combustion engines. To avoid excessive losses, the high density of the Cp-34 vapor at engine inlet conditions necessitates an inlet valve comparable in diameter and lift (and therefore mass) to the intake valve of an internal combustion engine. However, the valve event is much shorter in the Rankine engine than in the internal combustion engine. The design point intake ratio of 13.7% corresponds to a valve event of 60° at most, whereas in internal combustion engines the inlet valve event is of the order of 120° or more. In a cam operated system, at a given speed, the acceleration and, therefore, the stress level are proportional to the lift divided by the square of the valve event, so that the cam stresses are much higher at a given engine speed for the vapor engine.

One way of overcoming this problem is shown in Figure 2.5. In this system, two concentric inlet valves in series are driven by two separate camshafts. Cam number 1 driving inlet valve 1 has fixed timing with respect to the crankshaft. Cam number 2 has variable timing with respect to the crankshaft, and the total valve event is determined by the overlap of the two valves. In this way, relatively long cam events can be used, giving reasonably sized camshafts. Figure 2.2 also shows that the mean valve opening area can be higher with this approach than with a single valve, which should compensate for the lower flow coefficient of the two valve system.

Other approaches to variable cut-off inlet valving are shown in Figures 2.6 and 2.7. These are both hydraulic devices. A directly-actuated hydraulic system is shown in Figure 2.6: a cam operated plunger pump operates a hydraulic column which acts on a stepped piston on the inlet valve stem. The pump plunger is constructed with a helical undercut so that its angular position in its bore determines its effective stroke, thus varying inlet valve duration. This system is quite similar to diesel engine injection systems.

Another hydraulic scheme is shown schematically in Figure 2.7. In this system a pump supplies high pressure oil at 2000 - 3000 psi to a rotary valve (shown as two valves for simplicity in Figure 2.7). The rotary valve supplies the high pressure oil to alternate sides of a piston connected to the inlet valve. Cut-off adjustment is obtained by moving the rotary valve axially in its housing.

2.2.2.2 Bearing Design. The engine bearing design is strongly influenced by the type of transmission used. If the engine is coupled directly to the drive shaft, as in many early steam cars, then journal bearings relying on hydrodynamic lubrication cannot be used; roller or ball bearings must be used, because of the high bearing loads which could occur at essentially zero rpm. On the other hand, if a conventional torque converter were used, bearing sizes, at least on the crankshaft, can be comparable to an internal combustion engine of the same bore, since the peak cylinder pressures are roughly the same and the inertia loading is lower on the Rankine engine (because of the 1000 ft/min limit on piston speed). The wrist pin bearing is more heavily loaded than in the conventional four stroke internal combustion engine because the load on the pin never reverses; in this respect, it is much like the wrist pin in a two-stroke engine. Any single clutch system can load the engine bearings fairly heavily when the engine is idling at 300 rpm and the clutch is engaged. At these conditions, with a maximum intake ratio of 80%, the bearing loading is such that bearing sizes associated with a two stroke diesel of the same bore would be barely adequate for the Rankine engine. A clutch with two forward speeds as well as a reduction in the maximum intake ratio would alleviate this situation.

2.2.3 Feedpump Design

The primary factors influencing the selection and design of the vapor-generator feedpump are that the pump must be positive displacement because of the high discharge pressure; the lubricity of thiophene is relatively poor and its liquid viscosity low; the pumping rate must be variable from basically zero to 15 gpm over a 800 - 2000 rpm range; and the pump must operate with low NPSH without cavitation, since the NPSH is provided only by subcooling of the liquid coming from the condenser.

The feedpump selected, illustrated in Figure 2.8, is a 5-cylinder piston pump driven by a wobble plate; its characteristics are summarized in Table 2.3. The selection of a piston pump was based on testing of several types of positive displacement pumps, including gear and vane-type pumps, at Thermo Electron Corporation. In general, high leakage rates (low volumetric efficiency) and high wear rates have been encountered for all pump types other than piston, which has given completely satisfactory performance. With the piston pump all bearing surfaces can be oil lubricated.

The variable pump rate is obtained by incorporation of variable displacement in the feedpump, permitting the pumping rate to be controlled at the desired rate regardless of feedpump speed. The method used to obtain variable displacement is similar to that used in diesel fuel-injection pumps, in which a ramp undercut is machined in the piston and connected to the suction side by a port in the cylinder wall. As long as the port is covered by the piston, pumping occurs on the discharge stroke. As the port is uncovered by the undercut, the fluid in the cylinder is bypassed to the suction side as the piston continues its discharge stroke. Rotation of the piston with the ramp undercut varies the point at which the port is uncovered, thus varying the effective displacement of the feedpump. Pumping rates from zero to maximum are obtained by 180° rotation of the piston. In the wobble-plate design illustrated, rotation of all five pistons is obtained by use of a gear which meshes with gear teeth in the piston skirts. The gear is rotated by means of a rack and pinion drive passing external to the pump through a rolling diaphragm hermetic seal. One-half inch of rack motion rotates all five pistons simultaneously 180°.

Spring-loaded poppet suction and discharge valves are used. The suction valve is constructed in the cylinder and is made as large as possible to minimize the pressure loss through the valve and the tendency for cavitation. The smaller discharge valve is located in the cylinder head. Common suction and discharge plenums for all five cylinders are incorporated in the housing castings.

The use of five cylinders was based on reducing pressure transients due to the flow variation from the piston pump. These transients must be maintained sufficiently small on the suction side of the pump so that the liquid pressure never falls below the vapor pressure of the subcooled liquid. A computer analysis of the pressure transient behavior indicated that a five-cylinder pump would be required to prevent cavitation with 20°F subcooling at the pump suction.

The pump drive could be either crank or wobble-plate. The wobble-plate drive was selected because of its compactness and easier packaging with the engine, its lower weight and vibration, its quieter operation at higher speeds, and its more convenient geometry for variable displacement incorporation.

2.2.4 Burner-Boiler Design

The burner-boiler design is based on the following requirements:

Reference Cycle Boiler Heat Transfer Rate	1.58×10^6 Btu/hr
Maximum Boiler Heat Transfer Rate	1.70×10^6 Btu/hr
Burner Design Maximum Heat Release Rate (HHV)	2.06×10^6
Burner Design Efficiency (HHV)	82.5%
Turndown Ratio	15/1

In Figures 2.9 and 2.10, the boiler tube bundle design is presented; Figure 2.11 shows a cross section through the burner. The factors considered in arriving at this design, in addition to heat transfer performance, were low materials cost, low volume, easy construction, and low combustion side pressure drop.

The combustion gases, at a temperature of ~ 3300°F, flow from the combustion chamber into the center of the tube bundle and radially outward through the tube bundle. The flow path of the organic through the tube bundle is illustrated in Figure 2.12. The organic first flows through stage 1, from which the combustion gases are exhausted; this provides the lowest organic temperatures in the boiler at the combustion gas outlet and a high boiler efficiency without air preheat. It is important that an extremely compact and efficient heat transfer surface be used in this stage to maximize the boiler efficiency with acceptable pressure drop on the combustion side. The organic next flows through the inner stage through which the combustion gases first flow, with a resultant high heat transfer coefficient. Because of the high gas temperature and extended surface on the combustion side, coupled with the high heat transfer coefficient on the organic side, a very high heat transfer rate can be obtained in the first stage. The organic next flows through the superheater coil, or stage 3. This stage is a bare tube coil, since the controlling thermal resistance is on the organic side and an extended heat transfer surface is not required on the tube.

The characteristics of the three boiler stages are given in Table 2.4; Figure 2.12 presents the calculated design point temperature and pressure profiles through the boiler. In the last or third stage, a matrix made of steel balls brazed together and to the tube is used.

This type of extended surface provides a very high heat transfer rate per unit volume and is amenable to mass production techniques.

An intermediate heat transfer fluid (water) is used, with double tube construction in the boiler tubes, to positively prohibit hot spots on the organic side of the boiler.(1) Stages 2 and 3 are connected to the same water reservoir which represents the high pressure side of the boiler. The water side of stage 1 is separate and represents the low pressure water side of the boiler.

The combustion chamber design, illustrated in Figure 2.11, is based on design parameters supplied by the Marquardt Corporation, derived from experimental testing of a 500,000 Btu/hr burner of similar design. A volumetric burning rate of 2.8×10^6 Btu/hr-ft^3-atm was used in sizing the burner. The burner is constructed integral with the boiler tube bundle, as illustrated in Figure 2.9. To reduce the pressure drop, two identical burners are used rather than one longer burner with the same combustion chamber diameter. The pressure drop at maximum firing rate for the two burner setup is 1.5" w.c.

While no pollution measurements are available on the full-scale burner design illustrated, Thermo Electron Corporation has completed measurements on a 1/9 scale burner (120,000 Btu/hr) with performance characteristics similar to the burner illustrated in the design. This burner is being used on a 5 hp system now on test at Thermo Electron Corporation. Figures 2.13 and 2.14 present the steady state emission levels from this burner as a function of excess air for burning rates of 105,000 Btu/hr and 50,000 Btu/hr, respectively. It is apparent that the emission levels are extremely low. To indicate the transient performance of the burner, the burner was oscillated between 50,000 and

105,000 Btu/hr burning rates with constant fuel-to-air ratio maintained; CO and unburned hydrocarbon emission levels were monitored continuously while a bag sample was collected throughout the run for NO measurement at the end. The results are indicated in Table 2.5. As indicated in a later section, use of these emission concentrations with the system performance gives gm/mile emission levels significantly less than projected 1980 federal limits.

2.2.5 Condenser

The condenser core is similar to a Ford radiator with louvered fins, except that the flattened tubes have a heavier wall (0.030" vs 0.005") and are constructed in one integral piece with partitions used to provide the desired vapor-side flow path. Copper fins with 2.5 mil thickness are used, and the tubing is made of carbon steel rather than brass as in the Ford radiator. The condenser design is illustrated in Figure 2.15, with the design point characteristics given in Table 2.6. The frontal area used in the design represents the maximum practical area for the 1969 Ford Fairlane with some rework of the front-end frame and grill.

2.2.6 Regenerator

The regenerator design conditions are illustrated in Figure 2.16 and the regenerator design in Figure 2.17. The design is based on obtaining a compact regenerator with geometry suitable for packaging directly above the engine in the engine compartment and with low pressure drops on both the liquid and vapor sides. On the vapor side, a brazed ball matrix extended surface with 1/16 inch ball diameter is used. The exchanger is divided into four parallel liquid circuits;

in each circuit, the vapor passes through four separate stages, permitting the exchanger to approach that of a pure counterflow exchanger.

2.2.7 Automatic Transmission

Automatic transmission designs for the system are based on tailored application of transmission types currently used in automotive or other vehicular applications, reducing the development effort and uncertainty for this component. The Ford Motor Company has supplied information on one type which uses a standard 12" diameter automotive torque converter coupled with modification of a standard manual transmission for forward, reverse, neutral and park control. The Dana Corporation has provided a two-speed slipping clutch design based on the type of transmissions they supply for off-the-road vehicles. Above ~ 5 mph, this transmission locks so the system functions as a direct-drive system, while still permitting the engine to idle at zero vehicle speed to drive the accessories. Either of the two transmissions is feasible, and additional study is required to determine which is preferable.

2.2.8 Controls

The primary control problem in the system is control of the burning rate and of the pumping rate to the monotube vapor generator to maintain boiler outlet pressure and temperature within specified limits over any type of transient encountered by the system. The control system is based on operating the boiler as close to quasi-steady state as possible over all transients; both burning rate and pumping rate are maintained as closely as possible to the values corresponding to the instantaneous vapor flow rate from the regenerator.

The feedpump control, used to control the pumping rate to maintain boiler outlet pressure, is simplified by the fact that the organic flow rate at any engine rpm is approximately linear with intake ratio. The feedpump and engine valving are thus operated as a unit directly by the accelerator pedal; a vernier control working from the boiler outlet pressure is used to reduce deviations from the design point and to eliminate any imbalance in the system automatically. A mechanical governor is used to limit the maximum intake ratio as a function of rpm and to govern engine speed at idle.

In order to maintain the burning rate at a value corresponding to the organic flow rate into the boiler, the burner control uses an orifice in the organic line to sense almost instantaneously any changes in the organic flow rate. This signal, along with a similar signal from an orifice in the fuel line, is used with a diaphragm controller to provide the proper fuel and air flow rates. If necessary, the fuel-to-air ratio can easily be varied as a function of turndown to minimize pollutant emissions at any burning rate. A vernier control operating from the boiler outlet temperature is used to reduce deviations from the design point value; at low power levels, when the organic flow is low, this temperature control becomes the primary control on the burner.

2.3 SYSTEM PERFORMANCE AND PACKAGING

The important decisions which have a strong influence on the overall system performance and cost relative to the internal combustion engine are the method of driving accessories at zero vehicle speed and the constant intake ratio engine valving. With respect to the first decision, two alternatives are feasible. An auxiliary, constant speed engine can be used to drive all accessories, permitting the engine to be coupled directly to the drive shaft (through a simple gear transmission for forward, reverse, neutral, park control). A more complex transmission can also be used, permitting the main propulsion engine to idle at zero vehicle speed so that all accessories can be directly driven by the main propulsion engine; this transmission is still much simpler than that required for the I. C. engine driven system.

It has been the conclusion of this study that the preferable approach is use of the main propulsion engine to drive all accessories. Even though the accessories must be larger when driven by the variable-speed main propulsion engine, the accessory designs are identical and the same number of parts must be processed; very little cost differential exists between the different sized components. Any cost reductions due to smaller accessory components are more than counterbalanced by the requirement for an additional engine of 15 - 20 hp to handle short-term accessory peak loads, with governor-throttle valve control. The system with all accessories driven by the main propulsion engine, therefore, seems preferable in terms of simplicity, cost, and packaging; this system has been selected as the optimum approach.

With respect to engine valving, a detailed analysis with computer modeling of the engine and boiler performance for all operating conditions has been carried out, comparing a system with constant intake ratio engine valving and throttle valve control and a system with variable intake ratio engine valving. The results are summarized in the performance maps presented in Figures 2.19 and 2.20

for the constant IR and variable IR (IR_{max} = 0.29) systems, respectively. Also presented in Figure 2.20 is the power increment obtained by going to a system with $(IR)_{max}$ = 0.8, the maximum practical value. It is apparent that only a relatively small increase in the system maximum power level is obtained with $(IR)_{max}$ = 0.8, since going to $(IR)_{max}$ = 0.8 requires a much larger feedpump as well as maximum condenser cooling air at a lower engine and vehicle speed; $(IR)_{max}$ = 0.29 represents the optimum intake ratio. Comparing these two performance maps for the same boiler and engine sizes, the following conclusions can be made: (a) The system with variable IR valving has a peak efficiency of 18.5% versus 15.0% for the system with constant IR valving. This 20 - 25% improvement in efficiency (or in mpg) occurs over a large power-speed region including the region of 20 - 40 hp and 600 - 1200 rpm, where the system would operate most of the time; and (b) The peak power for the equivalent sized engine and boiler is much greater with variable IR valving than for constant IR valving. Calculations of 0 - 60 mph wide-open-throttle times indicate a 65% increase for constant IR valving relative to variable IR valving with $(IR)_{max}$ = 0.8. For these two reasons, there exists an extremely strong incentive for incorporation of variable intake ratio valving in the engine; this type of valving is used in the reference engine design presented earlier.

With reference to Figure 2.20, an important consideration is the part-load performance of the system. Thus, while the design point efficiency, defined by the peak power requirements for acceleration, is 13.7%, it increases under part load conditions where an automobile normally operates. This increase with the variable IR valving occurs because part-load operation is obtained by reducing the IR below the design point value of 0.137, providing a more efficient expansion in the engine; reduction in the condenser pressure under part-load operation also occurs again leading to a more efficient cycle. It will also be noticed that the region of high efficiency (>17%) is broad; that is, a high efficiency is obtained over a broad range of engine power and speed. Thus, while the peak thermal efficiency of the Rankine-cycle system (18.5%) is much less than that of the I. C. automotive engine (~ 30%), the average efficiencies for typical consumer driving cycles are much closer. The Ford Motor Company has calculated the fuel economies for different driving conditions of the Rankine-cycle system installed in a 1969 Ford Fairlane 4-door sedan using the performance map of Figure 2.20 with $(IR)_{max} = 0.8$ and compared them directly with the fuel economy calculated for the same body with 302 CID engine and three-speed automatic transmission. The results are summarized in Table 2.7 for both steady speed and dynamic driving cycle operation. For the customer average driving cycle, the mpg for the 302 CID I. C. engine is 15.7, versus 12.7 mpg for the Rankine-cycle system. With respect to acceleration performance, the Rankine-cycle system tractive effort with single-speed direct clutch transmission and with torque converter, is compared for $(IR)_{max} = 0.8$, in Figure 2.21, to that for the 302 CID I.C. engine with 3-speed automatic transmission and 2.79 axle ratio (calculations prepared by Ford Motor Company). At speeds above 43 mph, the tractive effort of the two systems is practically identical. Below this speed, the tractive effort from the I. C. engine is somewhat greater. The 0 - 60 mph WOT acceleration times are compared in Table 2.7 for the two systems (11.9 seconds for the I. C. system, compared to 14.2 seconds for the Rankine-cycle system with 184 CID engine).

In Figure 2.21, the tractive effort obtained by use of the Rankine-cycle system with $(IR)_{max} = 0.29$ and two-speed clutch transmission as calculated by the Dana Corporation is compared with that of the 302 CID I. C. engine with three-speed transmission. The tractive effort of the two systems is practically identical down to 20 mph. The 0 - 60 mph acceleration time for the Rankine-cycle system is reduced to 12.5 sec. for the combination, which appears optimum from an overall system viewpoint.

In Table 2.8, the weight of the Rankine-cycle system reference design (917 lbs total) is compared with that of the 302 CID engine with three-speed transmission (709 lbs total).

Packaging of the system is an extremely important consideration, particularly for the first generation prototype or production units when it is preferable to use as fully as possible the same car body as is used for I. C. powered autos. Accordingly, the approach followed has been to package the system completely in the engine compartment of current automobiles; a 1969 Ford Fairlane has been used for the current study, and a full-size, complete mockup of the system has been constructed in the engine compartment of this car with excellent results. In Figure 2.22, a photograph of the mockup is presented; in Figure 2.23, sketches illustrating the engine-transmission, burner-boiler, and condenser locations in the system are illustrated. The only change required in the engine compartment in packaging the system was modification of the frame and fender panels at the very front of the car to facilitate placement of the condenser.

2.4 EMISSION PROJECTIONS FOR RANKINE-CYCLE AUTOMOTIVE PROPULSION SYSTEM

Using the burner emission levels obtained from the burner developed at Thermo Electron Corporation for a 5 hp Rankine-cycle currently on test, projections have been made of the emission level of unburned hydrocarbons, CO, and NO from a Rankine-cycle automotive propulsion system using 10 mpg fuel economy; the results are presented in Table 2.9 and compared with projected Federal standards for 1975 and 1980; presented also are results for an uncontrolled I. C. engine and the IIEC targets for emission control for the I. C. engine. The Rankine-cycle system emission levels are lower than the projected 1980 standards by a factor of 5 for unburned hydrocarbons, by a factor of 13 for carbon monoxide emission, and by a factor of 1.6 for NO emission.

2.5 MAJOR CONCLUSIONS

1. The Rankine-cycle system offers the greatest potential of any combustion-operated propulsion system for absolute minimum emissions of particulates, unburned hydrocarbons, nitric oxide, and carbon monoxide.

2. The use of thiophene as a working fluid, with a moderate maximum cycle temperature of 550°F, permits a significant cost reduction relative to the equivalent steam system. This reduction may permit the Rankine-cycle system to be competitive costwise with the equivalent I. C. system, particularly since the stricter emission level standards will require significant cost increases in the I. C. system.

3. An organic working fluid, Rankine-cycle system equivalent in performance to a 302 CID I. C. engine with 3-speed automatic transmission can be packaged in current automotive engine compartments with only minor modifications required in the sheet metal and frame.

Reference

1. D. T. Morgan, E. F. Doyle and S. S. Kitrilakis, "Organic Rankine Cycle with Reciprocating Engine," Presented at the Fourth Intersociety Energy Conversion Engineering Conference, Washington, D. C., Sept. 22 - 26, 1969. Paper No. 699001.

TABLE 2.1

DESIGN POINT SPECIFICATIONS

Working Fluid	Cp-34
Boiler Outlet Temperature	550°F
Boiler Outlet Pressure	500 psia
Boiler Heat Transfer Rate	1.58×10^6 Btu/hr
Boiler Efficiency (HHV)	82.5%
Engine Displacement	184 in^3
Engine Speed	2000 rpm
Engine Piston Speed	1000 ft/min
Engine Horsepower Less Feedpump Power	103.2 hp
Engine Thermal Efficiency	86.0%
Engine Mechanical Efficiency	91.5%
Engine Overall Efficiency	77.5%
Engine IMEP	127.4 psi
Regenerator Effectiveness	90.0%
Regenerator Heat Transfer Rate	0.249×10^6 Btu/hr
Condensing Temperature	210.2°F
Condensing Pressure	25.0 psia
Subcooled Liquid Temperature	196.2°F
Condenser Heat Transfer Rate	1.25×10^6 Btu/hr
Organic Mass Flow Rate	7827 pounds/hr
Organic Volumetric Flow Rate	15.1 gallons/min
Feedpump Overall Efficiency	59.7%
Feedpump Power	5.25 hp
Cycle Efficiency	16.7%
Overall Efficiency	13.7%

TABLE 2.2

ENGINE DIMENSIONS AND SPECIFICATIONS

Configuration	Four Cylinders, 90°V
Bore	4.42 inches
Stroke	3.0 inches
Displacement	184 in^3
BHP (feedpump work deducted)	103 at 2000 rpm
IMEP at Design	127 psi
BMEP at Design	117 psi

TABLE 2.3

CHARACTERISTICS OF FEEDPUMP WITH 15 GPM RATING

(Sized for System with $(IR)_{max} = 0.29$)

Number of cylinders	5
Volumetric Efficiency	90%
Overall Efficiency	80%
RPM Range for Maximum Pumping Rate	800 - 2000 rpm
Total Displacement	4.78 in^3
Bore	1.595 in.
Stroke	0.478 in.
Materials of Construction	
Housing	Cast Iron
Pistons and Valves	Hardened Steel
Bearings	Roller and Needle

TABLE 2.4

BOILER REFERENCE DESIGN POINT SPECIFICATION

Stage No.	Heat Transfer Rate Btu/hr	Combustion Gas Temp. °F		Tubing Length	Pressure Drop	
		entering	leaving	ft.	Combustion Side in w.c.	Organic Side psi
2	0.834×10^6	3330	1896	17.0	0.136	21.2
3	0.383×10^6	1896	1190	35.0	0.288	23.3
1	0.359×10^6	1190	490	26.0	2.06	2.0
Total	1.576×10^6	–	–	78.0	2.48	46.5

Tube Specifications

Inner Tube ID = 0.930"
OD = 1.000"

Outer Tube ID = 1.125"
OD = 1.315"

External Fin Specifications

Fins/Inch	10
Fin Thickness	0.012"
Fin Material	Copper
Fin Height	0.356"

Matrix Specifications

Ball Size	3/32"
Ball Material	Carbon Steel
Matrix Thickness	0.5"
Matrix Height (between tubes)	0.935"

TABLE 2.5

TRANSIENT EMISSION DATA

FIRING RATE (BTU/HR)	EXCESS AIR (%)	CH_x (PPM)	CO (PPM)	NO_x (PPM)	ELAPSED TIME (MIN)
105,000	25	6	60	-	0
90,000	25	-	-	-	.5
90,000	25	5	30	-	1.0
90,000	25	-	25	-	1.5
90,000	25	4.5	70		2.0
105,000	25	40^1	180^1	-	2.5
105,000	25	7	90	-	3.0
105,000	25	5	-	-	3.5
90,000	25	5	80	-	4.0
90,000	25	4	25	-	4.5
50,000	25	4.5	20	-	5.0
90,000	25	-	-	-	5.5
105,000	25	15^1	$1000+^1$	-	6.0
105,000	25	-	-	-	6.5
105,000	25	6	75	-	7.0
90,000	25	-	75	-	7.5
90,000	25	4.5	30	-	8.0
90,000	25	4	15	-	9.0
				42^2	
				33^2	

NOTES:

1. Short duration peak ~ 10 sec.
2. Exhaust gas sample collected during the 9 minute run. Then two samples were drawn and NO analysis performed.

TABLE 2.6

CONDENSER DESIGN POINT CHARACTERISTICS

Heat Rejection Rate (20° Superheat Entering 20° Subcooling Leaving)	1.25×10^6 Btu/hr
Length	50 in.
Height	19.9 in.
Depth	3.0 in.
Frontal Area	6.91 ft^2
Condenser Inlet Pressure	25 psia
Design Ambient Air Temperature	95°F
Air Pressure Drop	3.45 in. w. c.
Shaft Fan Power (Two 20 in. O.D. Fans with 3.50 inch Pitch)	7.91 hp
Air Flow Rate	63,000 lb/hr

TABLE 2.7

PERFORMANCE COMPARISON OF RANKINE-CYCLE AND INTERNAL COMBUSTION AUTOMOTIVE PROPULSION SYSTEMS

Vehicle - 1969 Ford Fairlane 4-Door Sedan

Weight - 3539 lbs

System	Top Speed mph	0 - 60 mph Acceleration Time, sec	Fuel Economy, mph					
			Steady Speed, mph			City	Suburban	Customer Average
			30	50	70			
Ford Production Engine								
302-2V Engine with Automatic Transmission	106	11.9	27.3	20.4	16.2	13.3	18.0	15.7
250-1V Engine with Automatic Transmission	–	15.6	27.0	22.9	17.6	12.9	19.6	16.3
Rankine-Cycle Systems								
184 CID Engine $(IR)_{max} = 0.8$, Single Speed Clutch Transmission	100	14.2	33.1	20.4	13.4	9.6	15.8	12.7
220 CID Engine $(IR)_{max} = 0.8$, Single Speed Clutch Transmission	106	11.4	–	–	–	–	–	–
184 CID Engine, $(IR)_{max} = 0.29$ Two Speed Clutch Transmission 2.0/1 and 1/1 Gear Ratios	100	12.5	–	–	–	–	–	–

TABLE 2.8

TABULATION OF TOTAL SYSTEM WEIGHT AND COMPARISON WITH 302-2V INTERNAL COMBUSTION SYSTEM WITH 3-SPEED TRANSMISSION

	Rankine Reference Design	302 Cu. In V-8 with 3-Speed Automatic
Engine Expander Assembly	220	
Feedpump	45	
Engine Subsystem	265 lbs	479 lbs
Transmission	135 lbs	159 lbs
Burner-Boiler	273 lbs	
Regenerator	54 lbs	
Condenser	115 lbs	
Radiator with fan, connectors, and water		54 lbs
Controls, Exhaust, Electrical System, Accessory Drives, and other Miscellaneous Components	75 lbs	114 lbs
Working Fluid and Lubricant	40 lbs	
Total	957 lbs	806 lbs

TABLE 2.9

EMISSION COMPARISON OF RANKINE CYCLE WITH INTERNAL COMBUSTION ENGINE

System		Projected 1975 Standards	Projected 1980 Standards	Uncontrolled I. C. Engine	IIEC Targets for I. C. Engine	TECO Projections for Rankine-Cycle Propulsion System
Emission	HC	0.5	0.25	5.5	0.82	0.05
Level	CO	11	4.7	24	7.1	0.35
gms/mi	NO	0.9	0.4	5.5	0.68	0.25

2464-D

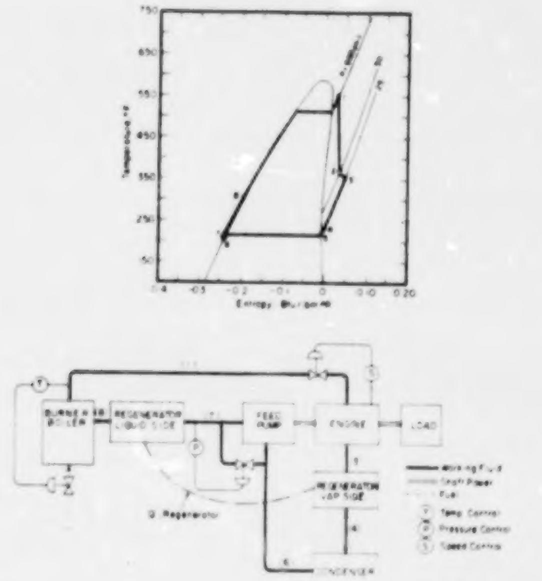

750
650
550
450
350
250
150
Temperature °F
Entropy Btu/lb°R
-0.2
-0.1
0
0.10
0.20
BURNER & BOILER
REGENERATOR LIQUID SIDE
FEED PUMP
ENGINE
LOAD
REGENERATOR VAP SIDE
CONDENSER
Q Regenerator
Working Fluid
Shaft Power
Fuel
Temp Control
Pressure Control
Speed Control

Figure 2.1 Flow Schematic and Cycle Conditions for Rankine Cycle with Thiophene Working Fluid.

1-314

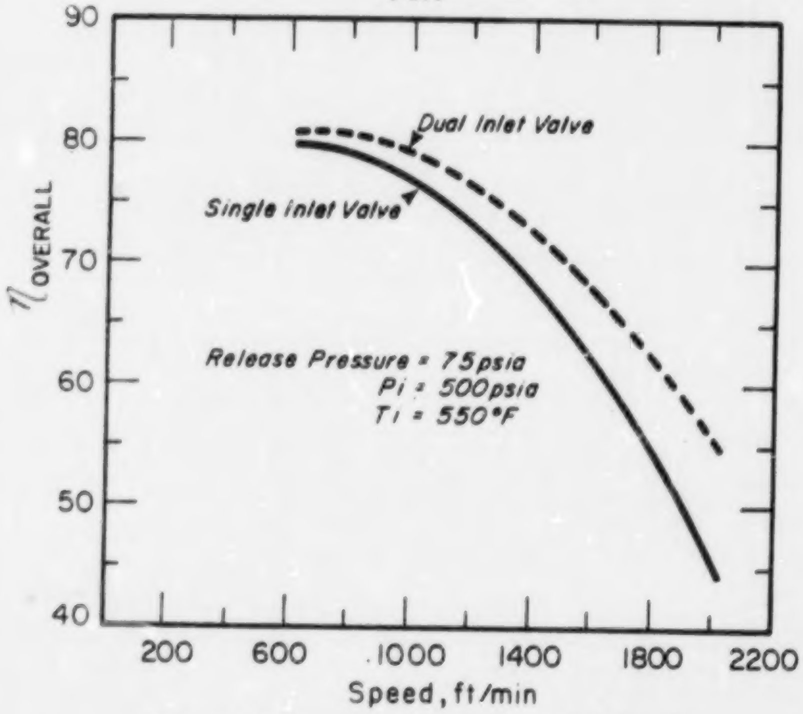


Figure 2.2 Overall Engine Efficiency Variation with Piston Speed.

2611-D

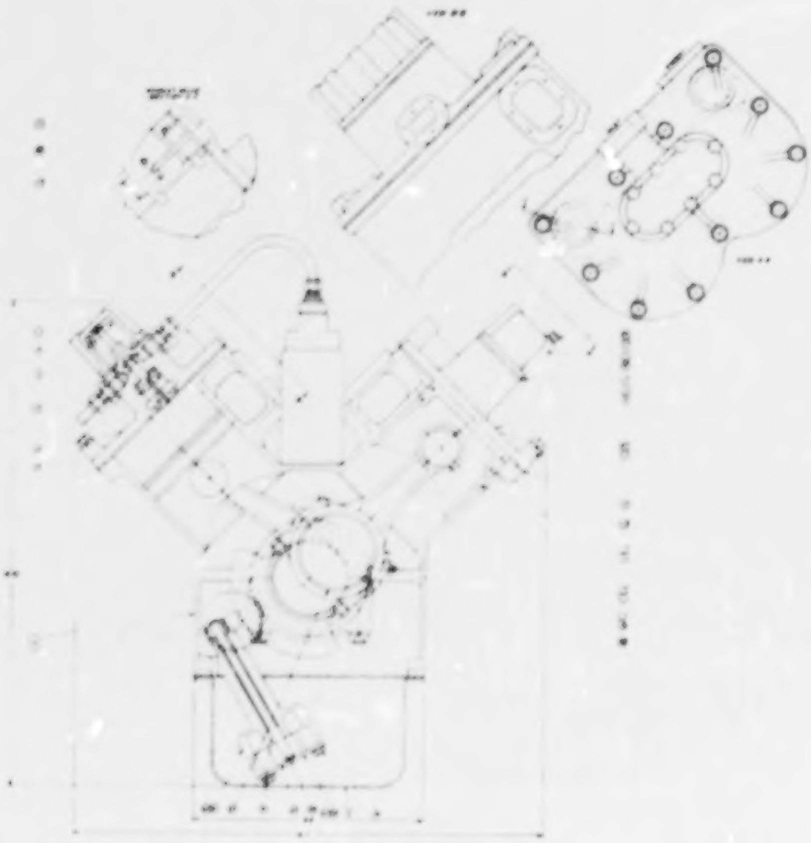


Figure 2.3 Cross-Sectional Drawing of Engine.

2612-D

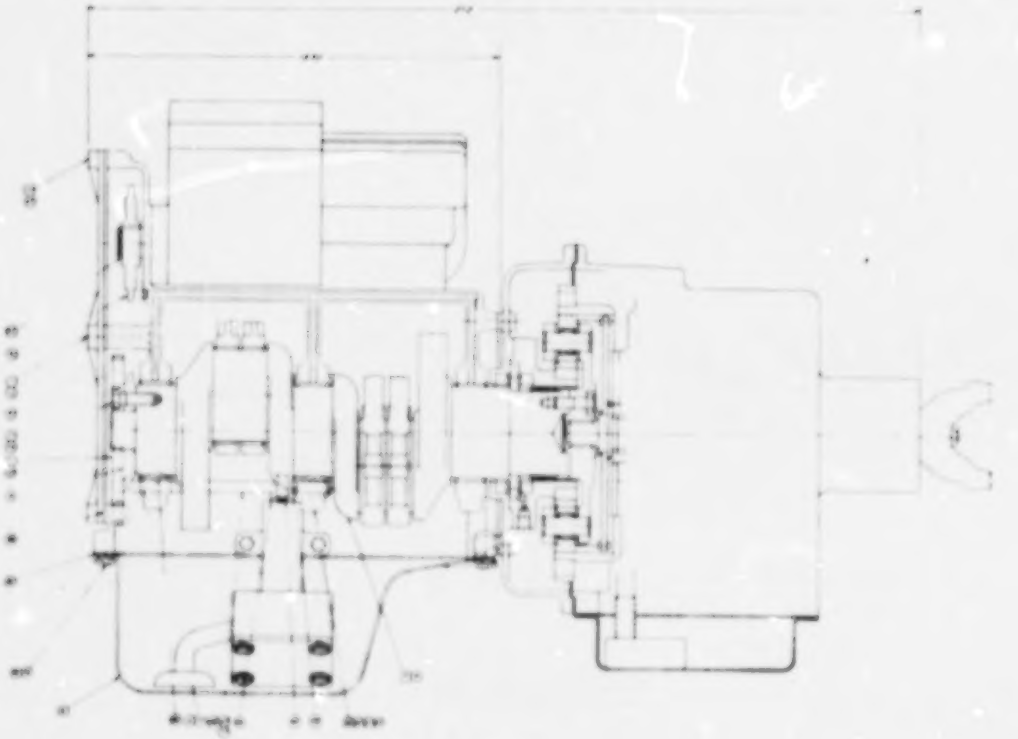


Figure 2.4 Longitudinal Section Drawing of Engine.

I-315

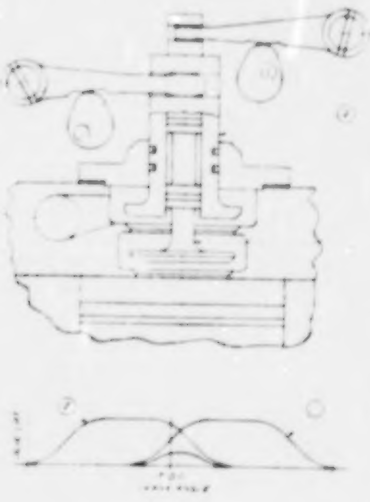


Figure 2.5 Two Inlet Valves in Series.

I-550

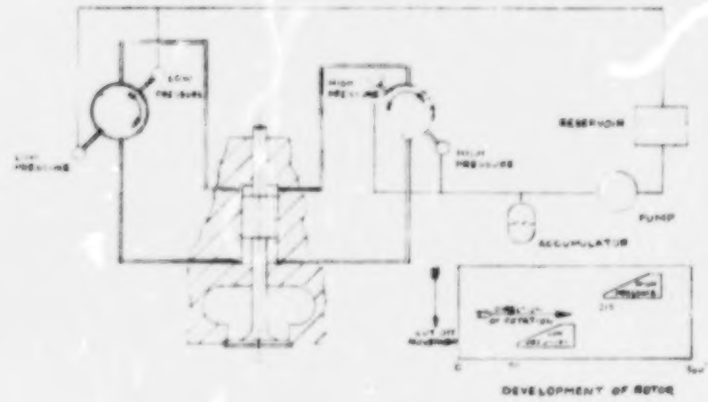

LOW PRESSURE
HIGH PRESSURE
RESERVOIR
PUMP
ACCUMULATOR
DEVELOPMENT OF ROTOR

Figure 2.6 Directly Actuated Hydraulic Valve.

I-551

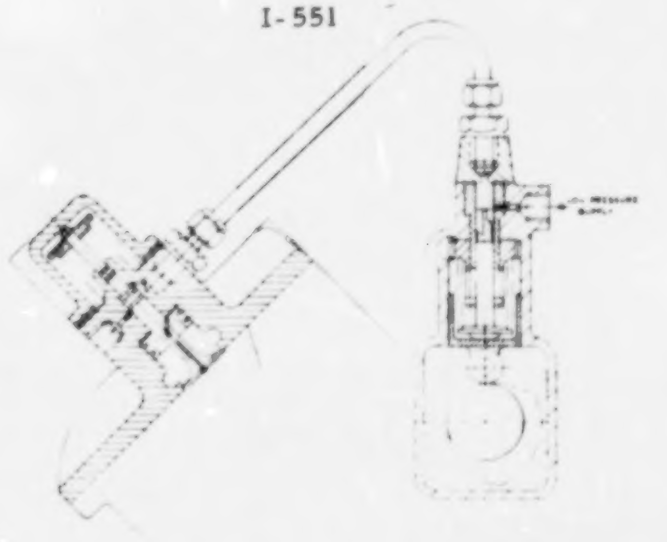


Figure 2.7 Pilot Operated Hydraulic Valve.

2623-D

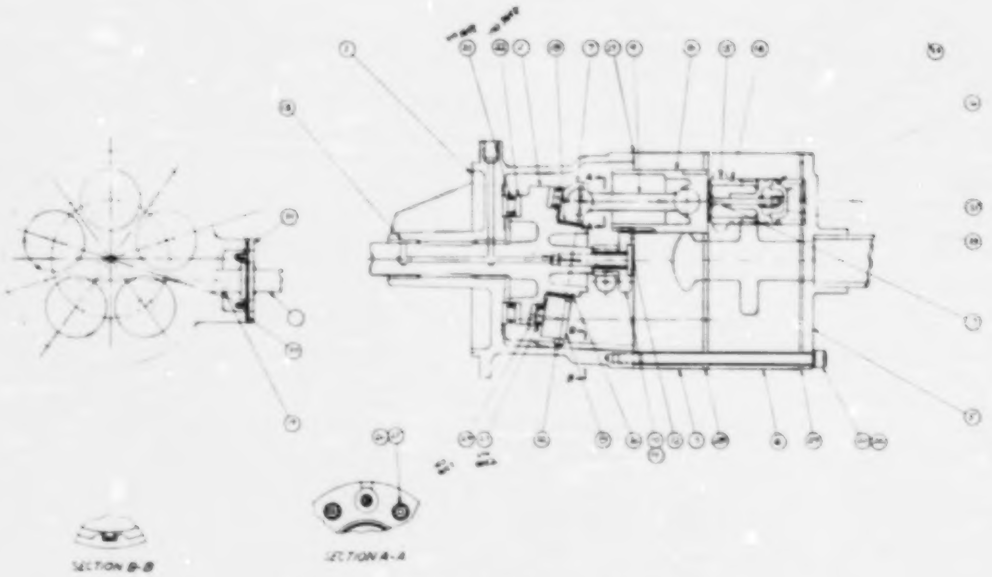

SECTION B-B
SECTION A-A

Figure 2.8 800-2000 rpm Feedpump.

2607-D

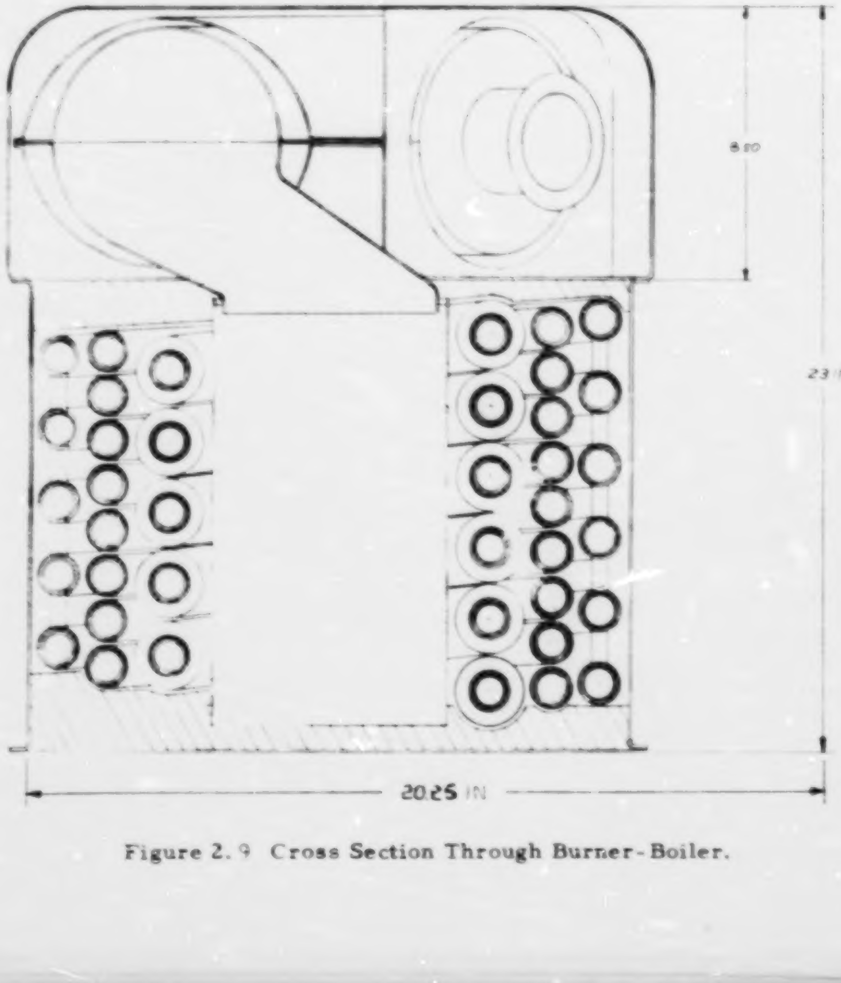

8.10
23 IN
20.25 IN

Figure 2.9 Cross Section Through Burner-Boiler.

2605-D

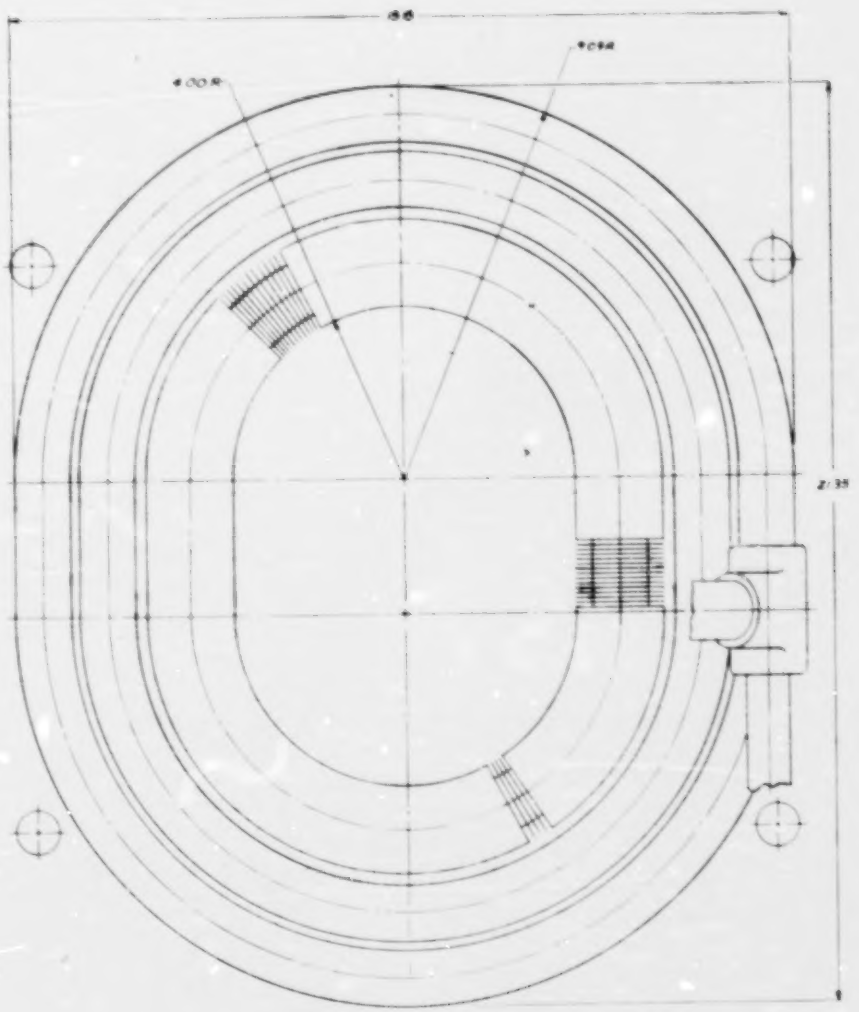

88
4.00 R
21.35

Figure 2.10. Top View of Boiler Tube Bundle.

2604-D

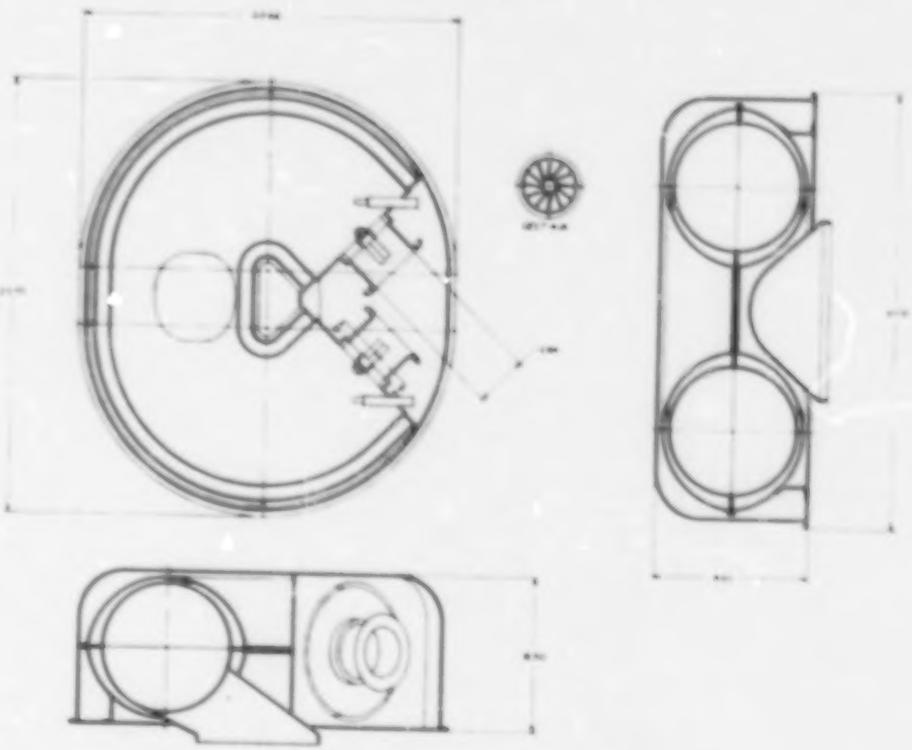


Figure 2.11 Cross Sections Through Automotive-Size Burner.

I-318

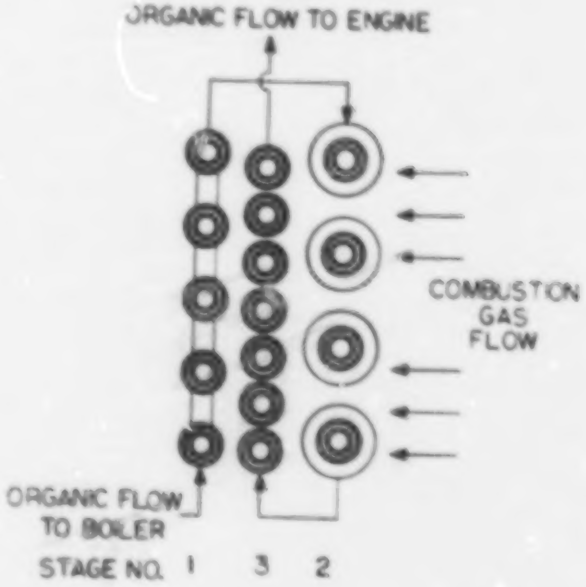

ORGANIC FLOW TO ENGINE
COMBUSTION GAS FLOW
ORGANIC FLOW TO BOILER
STAGE NO. 1 3 2

Figure 2.12 Organic Flow Path Through Boiler Tube Bundle.

I-916

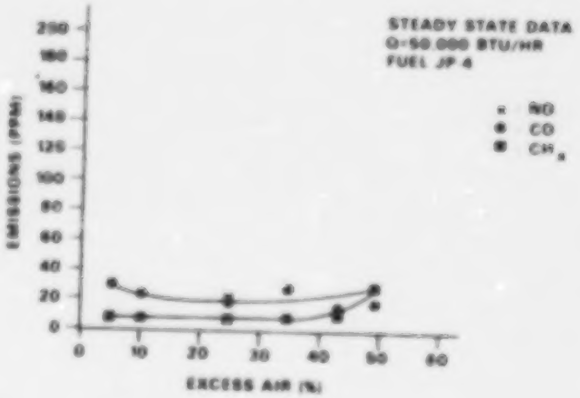


Figure 2.13 Effect of Excess Air on Emissions, 50,000 Btu/hr.

I-917

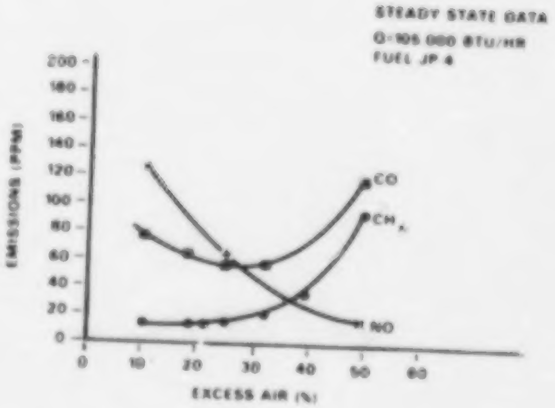


Figure 2.14 Effect of Excess Air on Emissions, 105,000 Btu/hr.

2615-D

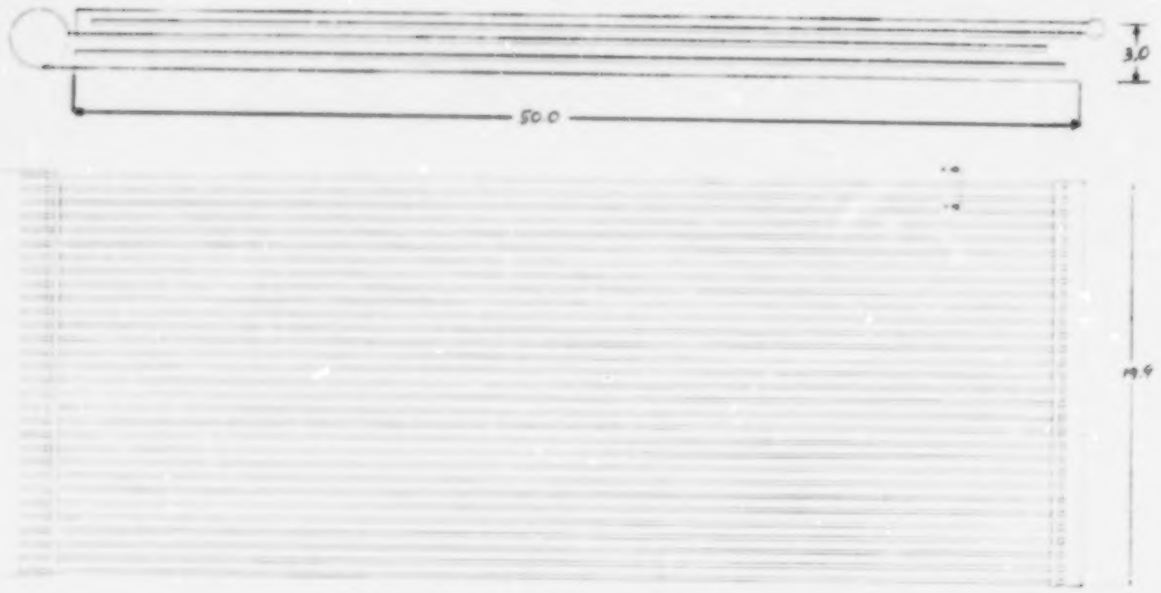


Figure 2.15 Condenser Design.

I-1060

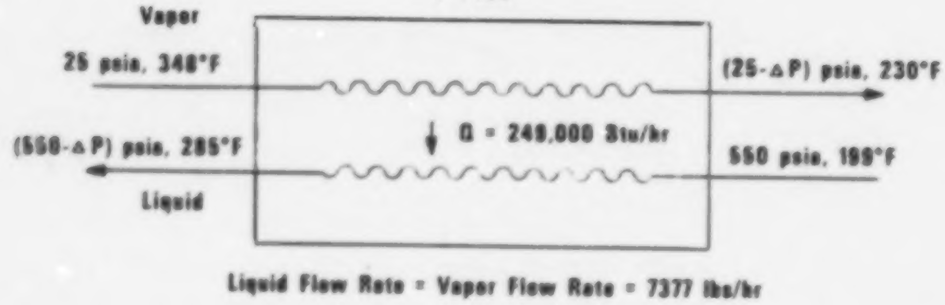

Vapor
25 psia, 348°F
(25-ΔP) psia, 230°F
Q = 248,000 Btu/hr
(550-ΔP) psia, 285°F
550 psia, 199°F
Liquid
Liquid Flow Rate = Vapor Flow Rate = 7377 lbs/hr

Figure 2.16 Regenerator Design Point Requirements.

2617-D

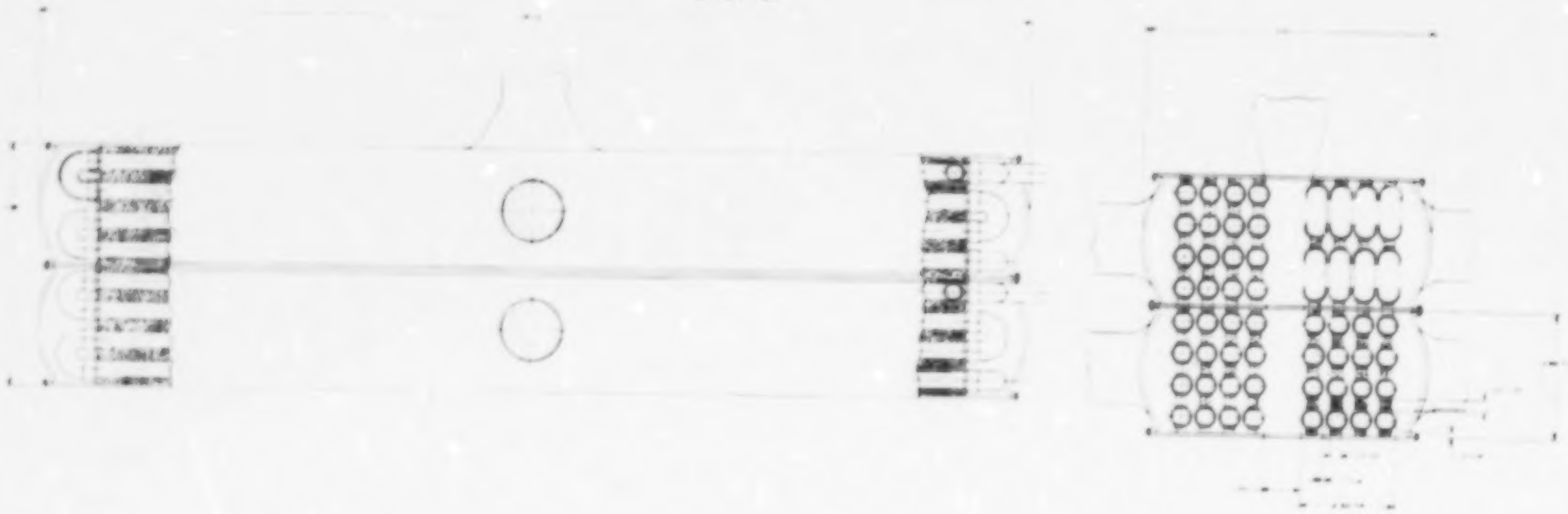


Figure 2.17 Regenerator Design.

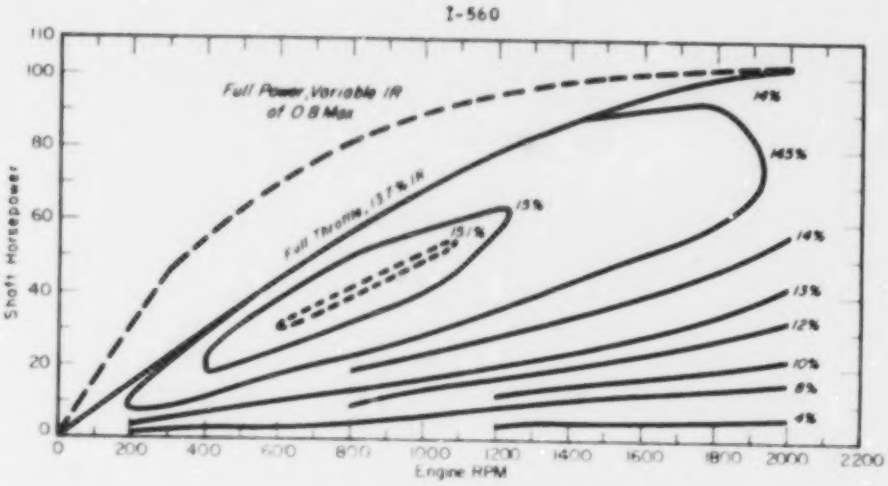


Figure 2.18 Performance Map with 184 CID Engine and Constant Intake Ratio of 0.137.

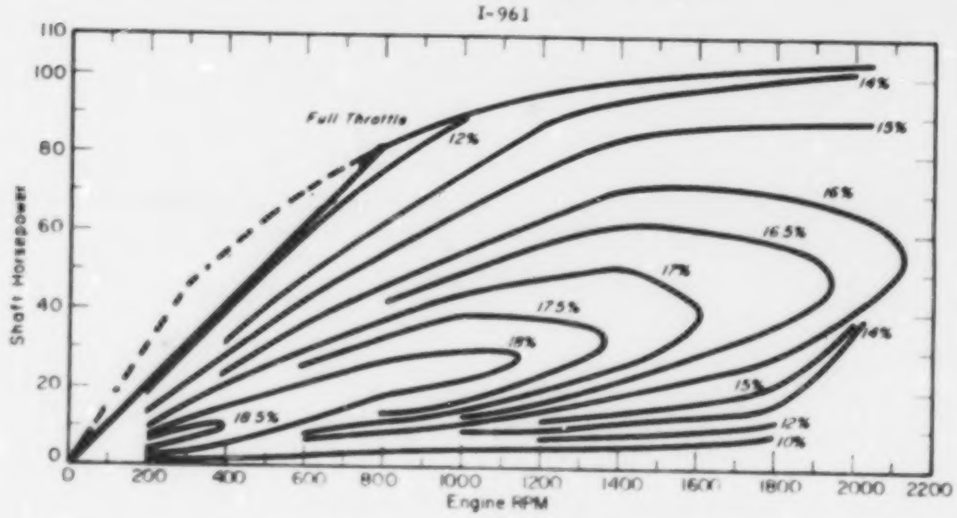
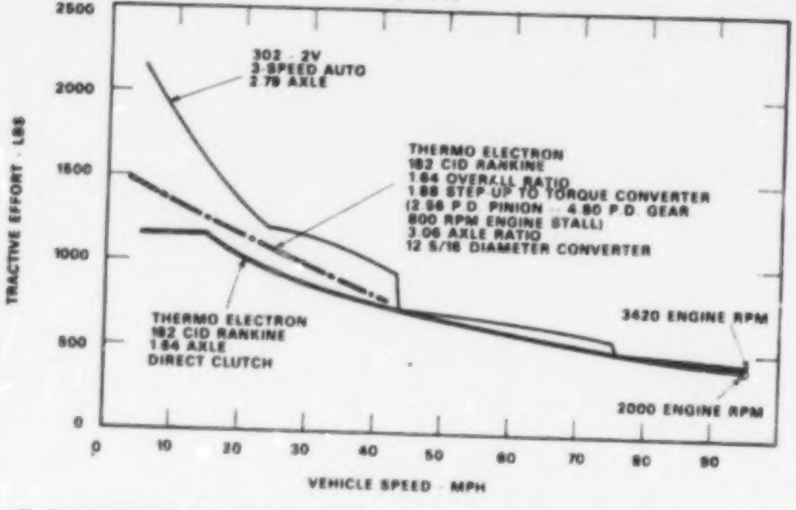


Figure 2.19 Performance Map with 184 CID Engine and Maximum Intake Ratio of 0.29.



NOTE Tractive effort computed by Ford based on engine performance data supplied by Thermo Electron for reference design system.

Figure 2.20 Tractive Effort versus Vehicle Speed, Fairlane at WOT Acceleration.

I-907

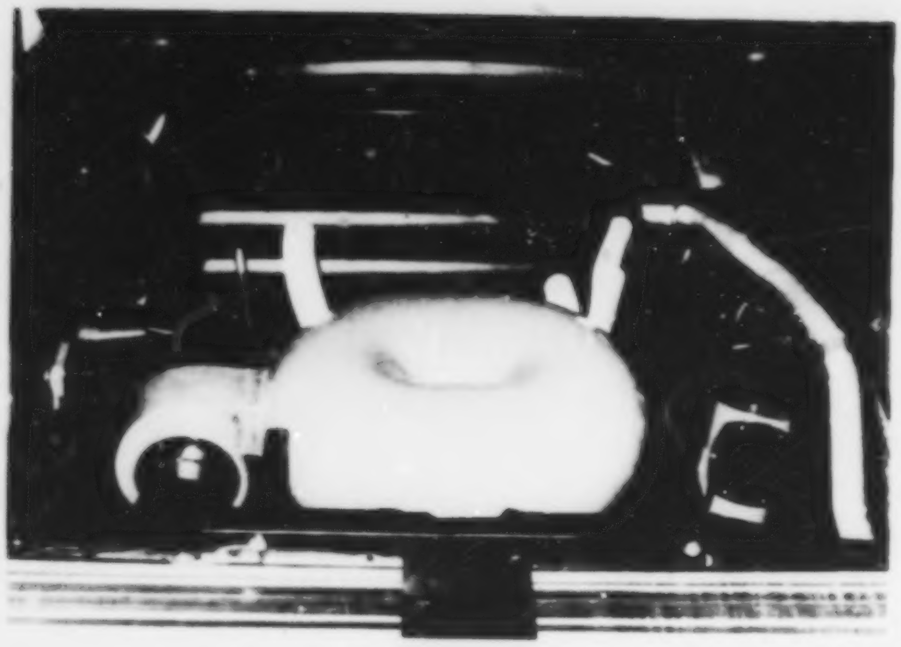


Figure 2.21 Photograph of Rankine-Cycle System Mockup in 1969 Ford Fairlane Engine Compartment.

I-970

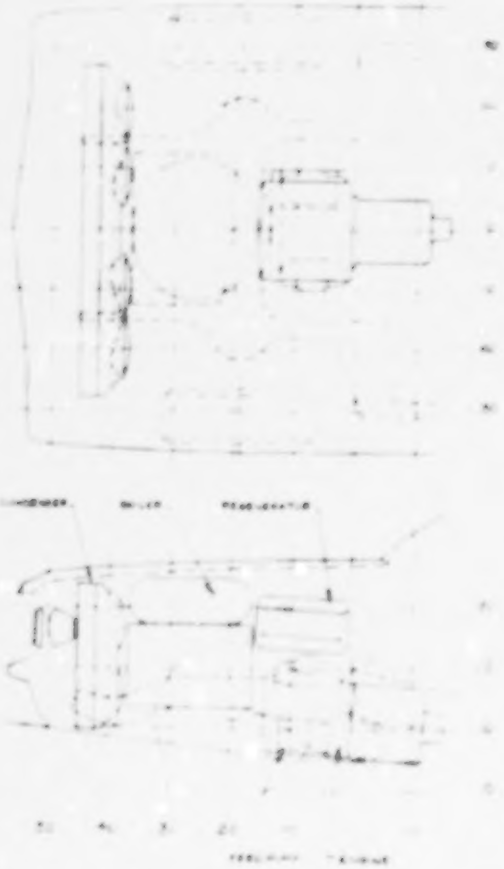

CONDENSER
BOILER
REGENERATOR

Figure 2.22 Rankine-Cycle Power Plant in 1969 Fairlane Engine Compartment

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3. INTRODUCTION

3.1 OVERALL GOALS

Great emphasis is being placed on air pollution reduction in the United States. Since the internal combustion engine powered automobile represents the largest contributor to air pollution, a large reduction in emission from this source represents an extremely important requirement for improving our air quality; strong federal and state pressure is being, and will continue to be, exerted on the automobile industry to reduce pollutant emissions.

The system which, comparatively, offers the greatest potential for absolute minimum emission of particulates, unburned hydrocarbons, NO, and CO, in a system with range and performance equivalent to the internal combustion engine, is the Rankine-cycle system. Quoting from a recent article by the Technical Advisory Committee, California Air Resources Board,[1]

> "The promise of the steam engine to produce inherently low levels of exhaust emissions is well-founded. The capability to utilize combustion processes which are not highly productive of the three principal pollutants is confirmed by available technical knowledge of flames and of the formation of those pollutants."

In Table 3.1, information prepared by the same Technical Advisory Committee is presented, summarizing their conclusions with respect to the emission levels which could be achieved by various approaches; the emission levels established either by legislation in California or as goals by various sources are also presented. It is apparent that the steam car, in the opinion of this Technical Advisory Committee, is low in all three of the major pollutants (hydrocarbons, carbon monoxide, and oxides of nitrogen), and that it is the only system which is simultaneously low in all three pollutants based on current technology.

Work at Thermo Electron Corporation in this study and on the hardware development of a 5 hp Rankine-cycle system, coupled with recent pollutant measurements performed by other industry groups from burners suitable for use in a Rankine-cycle system, have confirmed this inherent potential of the Rankine-cycle system. In Table 3.2, the projected pollutant emission levels using the results of three different types of burners are compared with current and projected Federal standards. It is apparent that emission levels for both hydrocarbons and CO are less than the projected 1980 Federal standards for all three burners. The NO level is less than the projected 1980 standard from the Thermo Electron Corporation measurements and slightly above the 1980 Federal standard from the GM and Marquardt measurements. These burners have not been optimized for reduction of NO emission, and it is expected that development work can reduce the NO emission to a level consistent with the projected 1980 standards with both HC and CO emissions being significantly lower than the 1980 standards.

With this general acceptance of the low pollution level of the Rankine-cycle system for automobile propulsion, the Motor Vehicle Research Division, National Air Pollution Control Administration, Public Health Service, Department of Health, Education, and Welfare, is supporting the development of such a system. A conceptual design of the system has been completed, detailed design and experimental development of the more critical components have been started.

TABLE 3.1

AUTO EMISSIONS: A SUMMARY OF POSSIBILITIES

(all figures in grams per mile)

Legislation and goals		Hydro-carbons	Carbon monoxide	Oxides of nitrogen
Prior to control		11.0	80.0	4.0
California Pure Air Act (AB 357)	1966	3.4	34.0	-
	1971	2.2	23.0	4.0
	1972	1.5	23.0	3.0
	1974	1.5	23.0	1.3
California Low Emission Vehicle Act (AB 356)		0.5	11.0	0.75
Morse report goals for 1975		0.6	12.0	1.0
Interindustry Emission Council goals		0.82	7.1	0.68
Modified conventional engines				
Sun Oil Co. test vehicle		0.7	12.0	0.6
Chrysler-Esso engines				
Manifold reactor		<1.5	<20.0	<1.3
Catalytic reactor		1.7	12.0	1.0
Synchrothermal reactor		0.25	7.0	0.6
Ethyl Corp. "lean reactor" car		<0.7	<10.4	<2.5
DuPont manifold reactor		0.2	12.0	1.2
Alternative power plants and fuels				
Steam car		0.2-0.7	1.0-4.0	0.15-0.4
Gas turbine		0.5-1.2	3.0-7.0	1.3 -5.2
Wankel engine		1.8	23.0	2.2
Stirling hybrid		0.006	0.3	2.2
Natural gas fuel		1.5	6.0	1.5

Source: Technical Advisors Committee, California Air Resources Board

TABLE 3. 2

PROJECTED EMISSION LEVELS FROM RANKINE CYCLE POWERED CAR ASSUMING 10 MPG AVERAGE FUEL ECONOMY

Projected Federal Standards, gms/mile						Source of Emission Data	Emission Levels					
1975			1980				ppm in Exhaust Gas			Calculated gms/mile for 10 MPG Fuel Economy		
HC	CO	NO	HC	CO	NO		HC	CO	NO	HC	CO	NO
0. 5	11	0. 9	0. 25	4. 7	0. 4	Measurements[(a)] at Thermo Electron Corp. on 140, 000 Btu/hr. Burner; Excess Air=33%	15	60	40	0. 05	0. 35	0. 25
						Measurements[(2)] by General Motors Corp., Excess Air = 68%	8	290	75	0. 033	2. 1	0. 59
						Measurements[(3)] by Marquardt Corp. on 500, 000 Btu/hr Burner, Excess Air = 33%	4	60	90	0. 013	0. 35	0. 57

(a) See Section 4. 5 for details of measurements.

The emphasis in this development is on reducing or eliminating the problems generally associated with Rankine-cycle automotive propulsion systems without compromising the emission or performance features of the system. The problem areas generally associated with the system which have been considered in the development of the design presented in this report are summarized in Table 3.3.

In addition to the goal of reducing the effect of, or eliminating, the problem areas outlined in Table 3.3, an additional guideline of the study has been the application of current state-of-the-art technology whereever possible in order to achieve the shortest possible development time for a prototype research vehicle. Areas where the current state-of-the-art is not applicable have been identified in the study so that the early hardware development can be concentrated in these areas. In the remainder of this section, a description will be given of the approach followed in accomplishing these goals. The other major sections of the report present the individual component characteristics and descriptions and the overall system performance and description for a 100 net shaft horsepower system competitive in performance with a 302 CID internal combustion engine coupled with a three-speed automatic transmission. The 1969 Ford Fairlane is used as the reference automobile chassis for the design.

3.2 APPROACH FOLLOWED FOR ATTAINMENT OF GOALS

The approach followed to alleviate the problems discussed above and to increase the competitiveness (in areas other than emission levels) of the Rankine-cycle system relative to the internal combustion system involves use of an organic working fluid with optimum thermo-

TABLE 3.3

PROBLEM AREAS GENERALLY ASSOCIATED WITH STEAM ENGINE DRIVE FOR AUTOMOBILES

1. High cost
2. Long startup time
3. Large package size and weight
4. Poor fuel economy
5. Safety problems
6. High maintenance and poor reliability
7. Freezing of working fluid at low ambient temperatures
8. Driving of accessories
9. Blowoff of working fluid under peak load conditions with makeup required

dynamic characteristics, such as thiophene, in place of water (which has generally been used in the past). In Figures 3.1 and 3.2, realistic cycle conditions suitable for use in a portable power system are presented on T-S diagrams for steam and thiophene, respectively. The cycle conditions selected for steam are based on the criterion that the expansion in the reciprocating engine should not enter the saturation dome, since this results in a large increase in heat transfer to the cylinder wall during the expansion, with a resulting decrease in the engine efficiency.

With respect to the steam cycle, the high pressure steam is expanded in the engine (process 1-3) to produce shaft power. The exhaust steam from the engine is then condensed and subcooled in the condenser (process 3-4). Water from the condenser is pumped back into the boiler by the feedpump (process 4-5). The burner-boiler then heats the water to the desired engine inlet conditions (process 5-1) completing the cycle.

An organic working fluid can be used just as easily as steam in a Rankine cycle, with the requirement for an additional component, a regenerative heat exchanger, and with precautions to prohibit overheating of the thermally-sensitive working fluid. The regenerator in an organic Rankine-cycle is desirable to increase the cycle efficiency, since the exhaust vapor from the engine for organic fluids is superheated and the energy in the exhaust vapor can be transferred to the feed organic going to the boiler, thereby reducing significantly the energy to be added to the working fluid in the vapor generator for a given power output from the engine. Because of its unique thermodynamic properties, thiophene requires a regenerator with a heat transfer rate relatively small compared to that of the vapor generator. Other organic fluids, particularly those with high molecular weight suitable for turbine expanders, may require a regenerator with a heat transfer rate close to or considerably in excess of the vapor generator heat input rate.

It is thus apparent that a Rankine-cycle system can be operated either with an organic working fluid or with steam as a working fluid. An organic fluid, however, offers a number of advantages over steam which have great significance for some of the problem areas associated with steam Rankine-cycle systems. A comparison of the principal characteristics of steam and organic working fluids is presented in Table 3.4. The advantages accrue from the much lower engine inlet temperature permissible with the organic working fluid without affecting the overall system efficiency. Thiophene, for example, gives a cycle efficiency with an engine inlet temperature of 550°F almost equal to that of the steam cycle with an engine inlet temperature of 800°F. This lower maximum temperature has several important benefits to a practical, low cost Rankine-cycle system with reciprocating engine. First, it permits use of less expensive materials of construction in the engine. Second, it eliminates the need for a reciprocating seal between the crankcase and power side of the engine to keep the lubricant separated from the working fluid. Lubricants are available which are thermally stable and compatible with the organic working fluid at the low maximum cycle temperature of 550°F. Lubricant entrained in the working fluid can, therefore, be allowed to pass through the vapor generator. Both of these features lead to a significant reduction in the cost of the reciprocating Rankine engine. In addition, elimination of the reciprocating seal leads to an increased

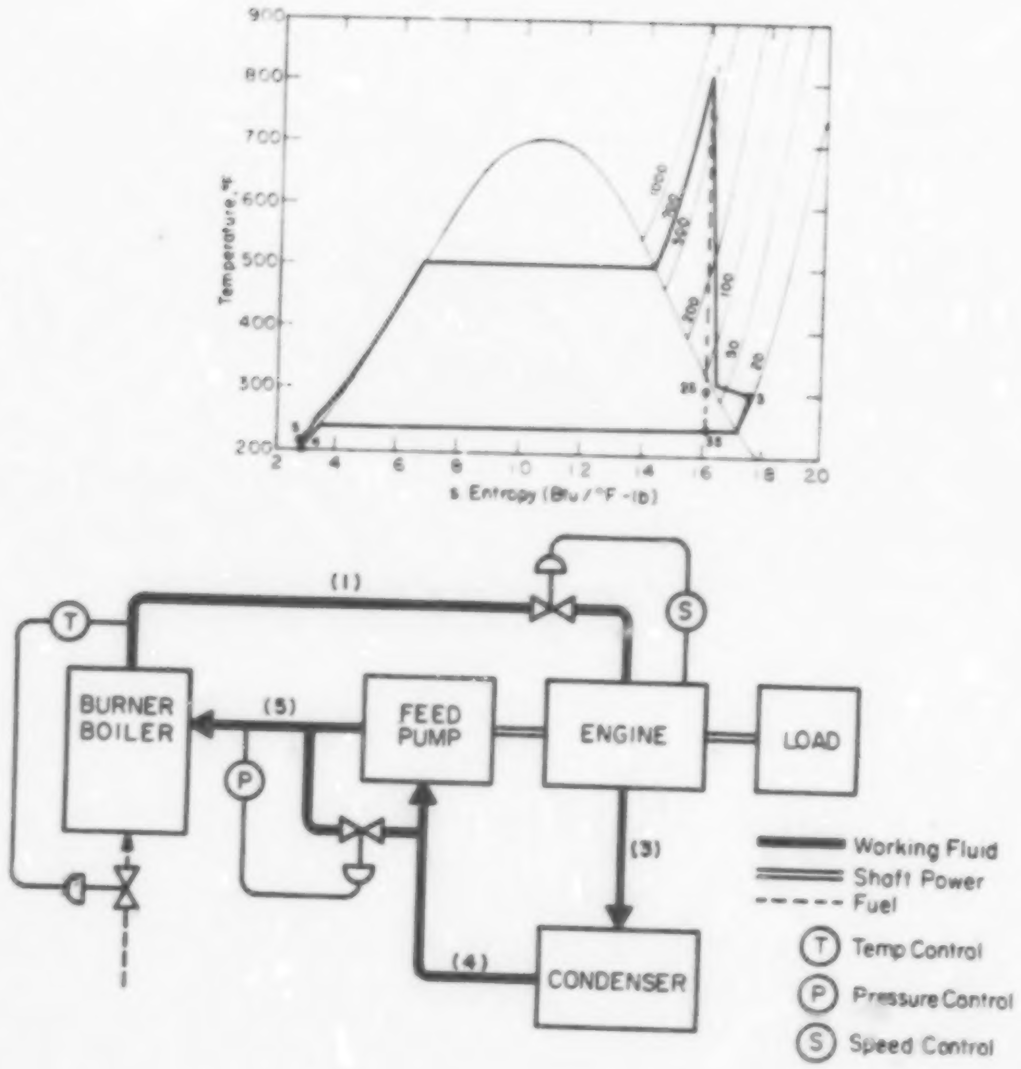

900
800
700
600
500
400
300
200
Temperature, °F
2
4
6
8
10
12
14
16
18
20
s Entropy (Btu / °F -lb)
1000
700
500
200
100
50
20
2S
3S
3
5
4
(1)
(5)
(3)
(4)
T
S
P
BURNER BOILER
FEED PUMP
ENGINE
LOAD
CONDENSER
Working Fluid
Shaft Power
Fuel
T Temp Control
P Pressure Control
S Speed Control

Figure 3.1 Flow Schematic and Cycle Conditions for Steam Rankine Cycle.

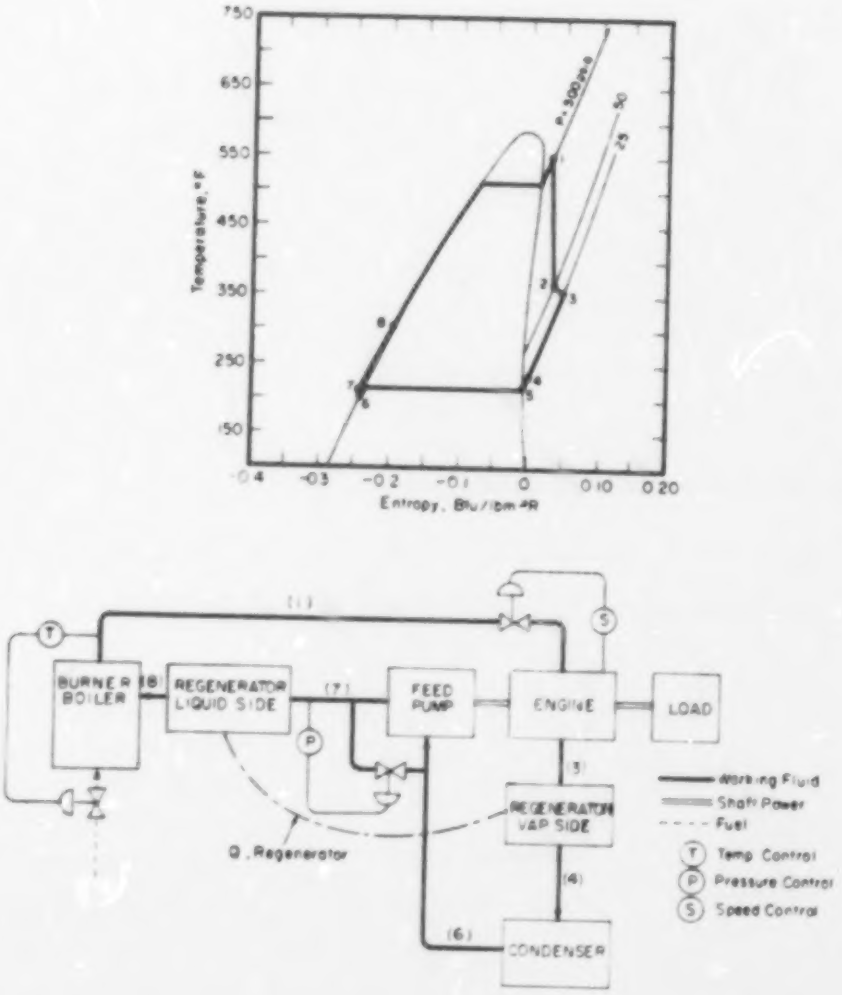

750
650
550
450
350
250
150
Temperature, °F
-0.4
-0.3
-0.2
-0.1
0
0.10
0.20
Entropy, Btu/lbm °R
P = 300 psia
50
25
1
2
3
4
5
6
7
8
(1)
BURNER BOILER
(8)
REGENERATOR LIQUID SIDE
(7)
FEED PUMP
ENGINE
LOAD
(3)
REGENERATOR VAP SIDE
(4)
(6)
CONDENSER
Q, Regenerator
T
P
S
Working Fluid
Shaft Power
Fuel
T Temp Control
P Pressure Control
S Speed Control

Figure 3.2 Flow Schematic and Cycle Conditions for Organic Rankine Cycle.

TABLE 3.4

COMPARISON OF STEAM AND THIOPHENE AS RANKINE-CYCLE WORKING FLUIDS IN SYSTEM WITH RECIPROCATING ENGINE

Thiophene Working Fluid	Water Working Fluid
- Low Operating Temperature (500°F) and Non-Corrosive	- High Operating Temperature and Corrosive
Conventional materials of construction (carbon steel, cast iron, aluminum, brass).	Alloy steels required in boiler and engine.
Compatible with lubricants at maximum cycle temperature.	Lubricant not compatible with working fluid at maximum cycle temperature.
- Lubricant Sealed in System with Working Fluid	- Poor Lubricating Properties
Bearings and sliding surfaces oil lubricated.	Water lubricated materials in engine.
Conventional engine construction (no reciprocating seal and crosshead piston).	Reciprocating seal in every cylinder.
	Crosshead piston required.
- Low Freezing Point (-40°F)	- High Freezing Point
- Limited Experience	- Extensive Experience
- Flammable/Toxic	- Non-Toxic/Non-Flammable
- Readily Available/Relatively High Fluid Cost	- Readily Available/Low Cost
- Thermal Decomposition (High Temperature Limit)	- Thermally Stable
- Regenerator Required	- No Regenerator Required

life without maintenance. The construction of the engine, in fact, is identical to that of internal combustion engines and reciprocating compressors. In Figure 3.3, a schematic comparison is given of the construction of an engine with and without a reciprocating seal.

The system is constructed as a completely sealed system with condenser capacity sufficient for peak load conditions. The only dynamic seal in the system is the rotary shaft seal on the rear of the engine, required for transmission of shaft power from the system. A seal has been tested at Thermo Electron Corporation which prohibits leakage of air into, or working fluid from, the system. The working fluid and lubricant are thus sealed in the system for the life of the unit. This approach is similar to that of a hermetically-sealed air conditioning system and should result, with development, in a system with high reliability and low maintenance requirements.

A very important problem with steam is its freezing point of 32°F; thiophene, on the other hand, has a freezing point of -40°F and can thus be used in all parts of the continental United States. Thiophene also contracts on freezing, so that exposure to temperatures less than -40°F will not damage the system.

In summary, use of thiophene as working fluid in a completely sealed Rankine-cycle system for automotive propulsion has the following advantages:

1. Minimum cost for complete system.
2. High reliability and low maintenance requirements.
3. Capable of cold startup down to -40°F ambient temperature.
4. Capable of storage at temperatures less than -40°F ambient temperature.

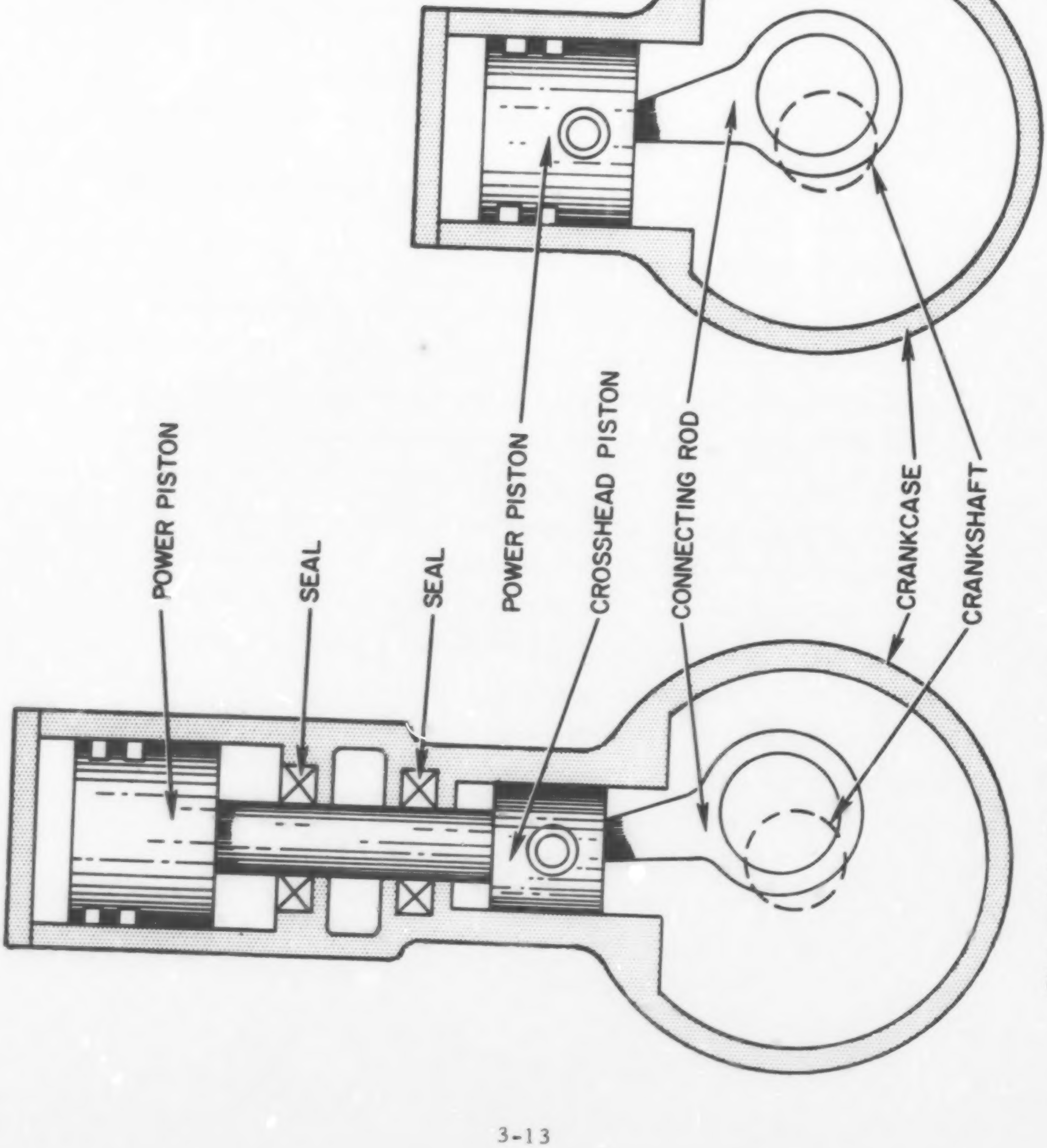

POWER PISTON
SEAL
SEAL
POWER PISTON
CROSSHEAD PISTON
CONNECTING ROD
CRANKCASE
CRANKSHAFT

Figure 3.3 Comparison of Conventional Steam Engine with TECO Engine Using Organic Working Fluid.

The component and overall system designs and characteristics presented in Sections 4 and 5 have established the capability for:

1. Startup time of 30 - 45 seconds.
2. Acceptable packaging in engine compartment of current automobiles.
3. Acceptable weight for current automobiles.
4. Customer average MPG 20% less than internal combustion engine.
5. Sufficient condenser capacity for totally condensing system under peak load conditions.
6. Conveniently driving all accessories.

One key difference between an organic and a steam Rankine cycle lies in the thermal stability of the working fluid. Water, of course, is completely stable under the conditions encountered in Rankine-cycle systems. Organic materials, on the other hand, have a definite and relatively sharp upper limit to the temperature to which they can be exposed without catastrophic thermal degradation of the fluid. To achieve a reasonable cycle efficiency, however, it is necessary to operate at a maximum cycle temperature close to that temperature limit. In any direct-fired boiler in which the heat is transferred directly from the combustion gases (at a maximum temperature of 3300°F) to the organic with only a metal wall between, hot spots will occur, leading to catastrophic decomposition of the organic working fluid unless very low peak cycle temperatures are used. At Thermo Electron Corporation, the approach followed has been to transfer the energy from the combustion gases to the organic by means of a thermally-stable, intermediate heat transfer fluid, thereby positively prohibiting exposure of the organic to excessive temperatures. This idea is not new, but its application has in the past required complicated and expensive equipment and has therefore been limited to large systems. The vaporizer concept developed at Thermo Electron, as described in Section 4, permits use of an intermediate heat transfer fluid to eliminate hot spots, with only a slight increase in cost over the equivalent direct-fired boiler; it is geometrically identical to and the same size as the equivalent direct-fired boiler, and requires no pump for circulation of the intermediate heat transfer fluid. This vapor generator concept has been tested with thiophene and with R-22 working fluids in two separate boiler loops which closely simulate the actual temperature conditions in a closed Rankine-cycle system for each fluid with no measurable degradation.

With respect to safety, use of a once-through boiler with a small high-pressure fluid inventory greatly alleviates any potential hazard from the relatively high boiler pressure. Use of an organic working fluid can introduce a significant hazard relative to water, however, because of the flammability and toxicity of the working fluid. While these hazards cannot be completely eliminated, proper mechanical design of the system coupled with design for minimum working fluid inventory should reduce the hazard to an acceptable level for the first experimental prototype. The safety considerations for use of thiophene working fluid are discussed in Section 4.2.

In selecting the working fluid for the first experimental prototype, greater emphasis has been placed on use of a fluid which provides a minimum cost system with excellent performance and efficiency characteristics than on minimum hazard. Once competitiveness with the internal combustion engine has been demonstrated in these critical areas, with an accompanying extremely low emissions level, a major effort on hazards evaluation can be justified, including experimentally simulated failures in the system as well as working fluid modifications to minimize any potential hazards. If competitiveness cannot be demonstrated, such an effort is not justified. Accordingly, the decision of working fluid was based primarily on selecting the existing working fluid which offered the best potential for fulfilling the economic, performance, and efficiency goals, particularly since the "hazard" of the system is a relative factor difficult to evaluate quantitatively in other than an experimental fashion. An additional factor in the selection of thiophene as the working fluid was the extensive experience gained at Thermo Electron Corporation in the development of a 5 hp system using thiophene as working fluid.

REFERENCES

1. "Post-1974 Auto Emissions: A Report from California," Environmental Science and Technology, pp. 288-294, Vol. 4, No. 4, April, 1970.
2. Vickers, P. T., et al., "The Design Features of the GM SE-101-A Vapor-Cycle Powerplant," SAE Paper 700163, January 12-16, 1970.
3. Personal Communication, April, 1970, Mr. Curtis Burkland, Marquardt Corporation, Van Nuys, California.

4. COMPONENT CHARACTERISTICS AND DESCRIPTIONS

4.1 INTRODUCTION

As with any other type of automotive propulsion system, the size of the system is set by the peak power or torque demands rather than by the average demands. The component sizes were based on a design power level of 100 net shaft horsepower at 2000 rpm engine speed, corresponding to a vehicle speed of 95 mph when directly coupled to the drive shaft. This maximum engine speed was based on a piston speed of 1000 ft/min, above which the intake valve loss increases rapidly. The engine for this power level is a V-4 with 184 cubic inch displacement.

The engine design is based on use of variable intake valving for control of the engine power output, thereby obtaining the maximum possible efficiency from the system under part-load conditions. This factor is of crucial importance for an automotive propulsion system, which operates at part-load most of the time with the excess power required only for acceleration performance. The design point intake ratio was taken as 0.137, based on a compromise between engine size, boiler-condenser size, and system efficiency. For example, for larger design point intake ratios, the engine size can be smaller for a given power output; the overall system efficiency is less, however, requiring a larger boiler-condenser size for a given power output. The part-load system efficiency and performance map is also dependent on the design point intake ratio. Increasing the design point intake ratio lowers the system horsepower at which the maximum efficiency occurs, while decreasing the design point intake ratio increases the system horsepower for maximum system efficiency. Thus, the design point intake ratio can be used within limits to tailor the performance map. For example, a lower design point intake ratio with a larger engine would provide a more efficient system than the reference design of this study for most driving conditions. In Figures 4.1.1 and 4.1.2, the performance maps calculated for a design point intake ratio of 0.137 at 2000 rpm are presented for maximum intake ratios of 0.8 and 0.29, respectively. The maximum intake ratio which can practically be used is 0.8.

The maximum intake ratio has a large influence on the feedpump displacement. In Figure 4.1.3, the maximum organic flow rates are presented for $(IR)_{max} = 0.8$ and $(IR)_{max} = 0.29$. For $(IR)_{max} = 0.8$, the feedpump must supply the maximum pumping rate down to 300 rpm; for $(IR)_{max} = 0.29$, down to 800 rpm; and for a fixed intake ratio with throttle valve control, the feedpump must supply the maximum pumping rate only at the design point speed of 2000 rpm. Feedpump designs have been prepared for all three approaches for size comparison.

In Table 4.1.1, the design point cycle conditions are presented as calculated with the system performance computer program, and in Table 4.1.2, a summary of the design point conditions is presented. The condenser fan and line pressure losses have not been included in the reference cycle design point calculations. With a condenser pressure of 25 psia and 82.5% boiler efficiency, based on the fuel higher heating value, the overall cycle efficiency is 13.7%. The design point engine release pressure is 58.8 psia.

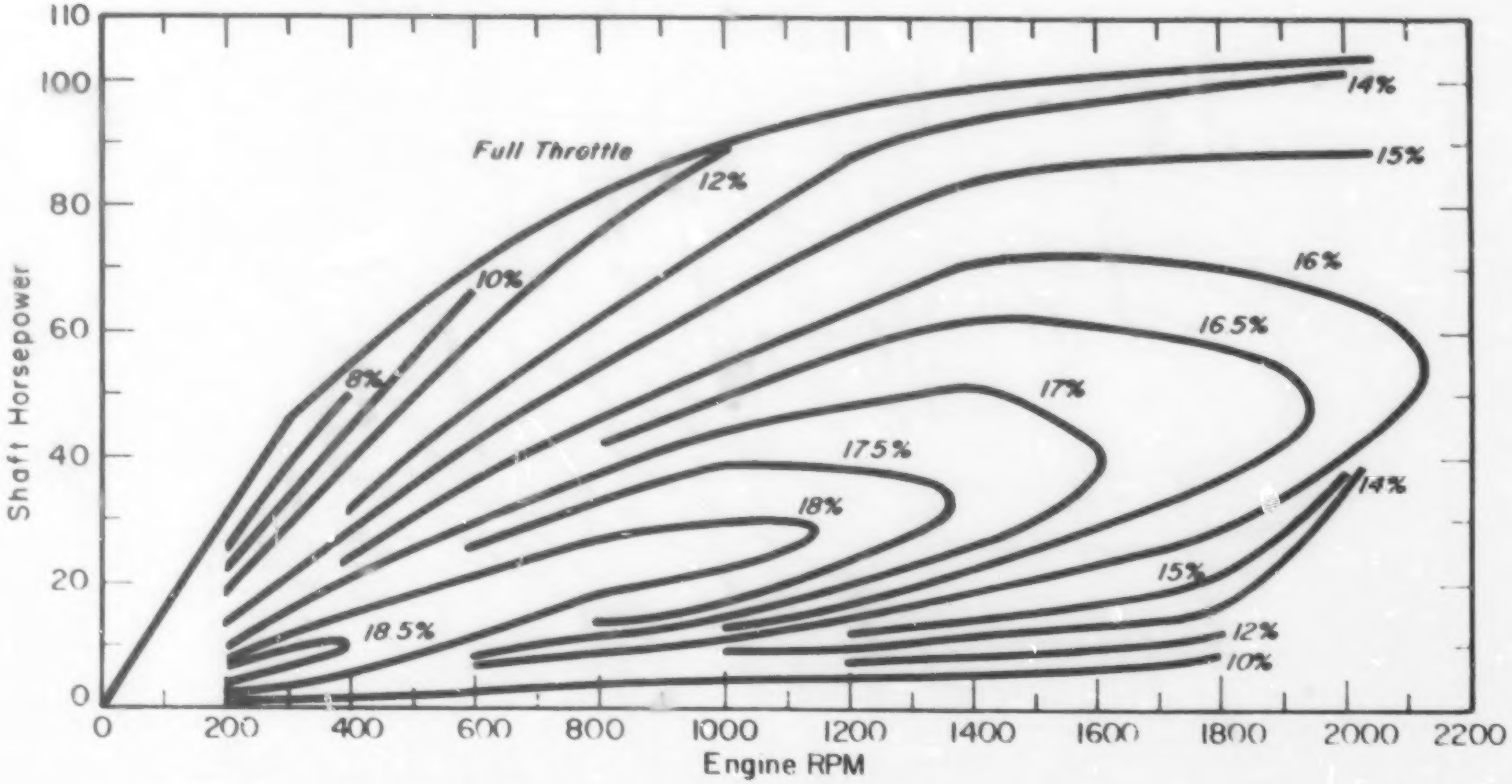


Figure 4.1.1 Performance Map with 164 CID Engine and Maximum Intake Ratio of 0.8.

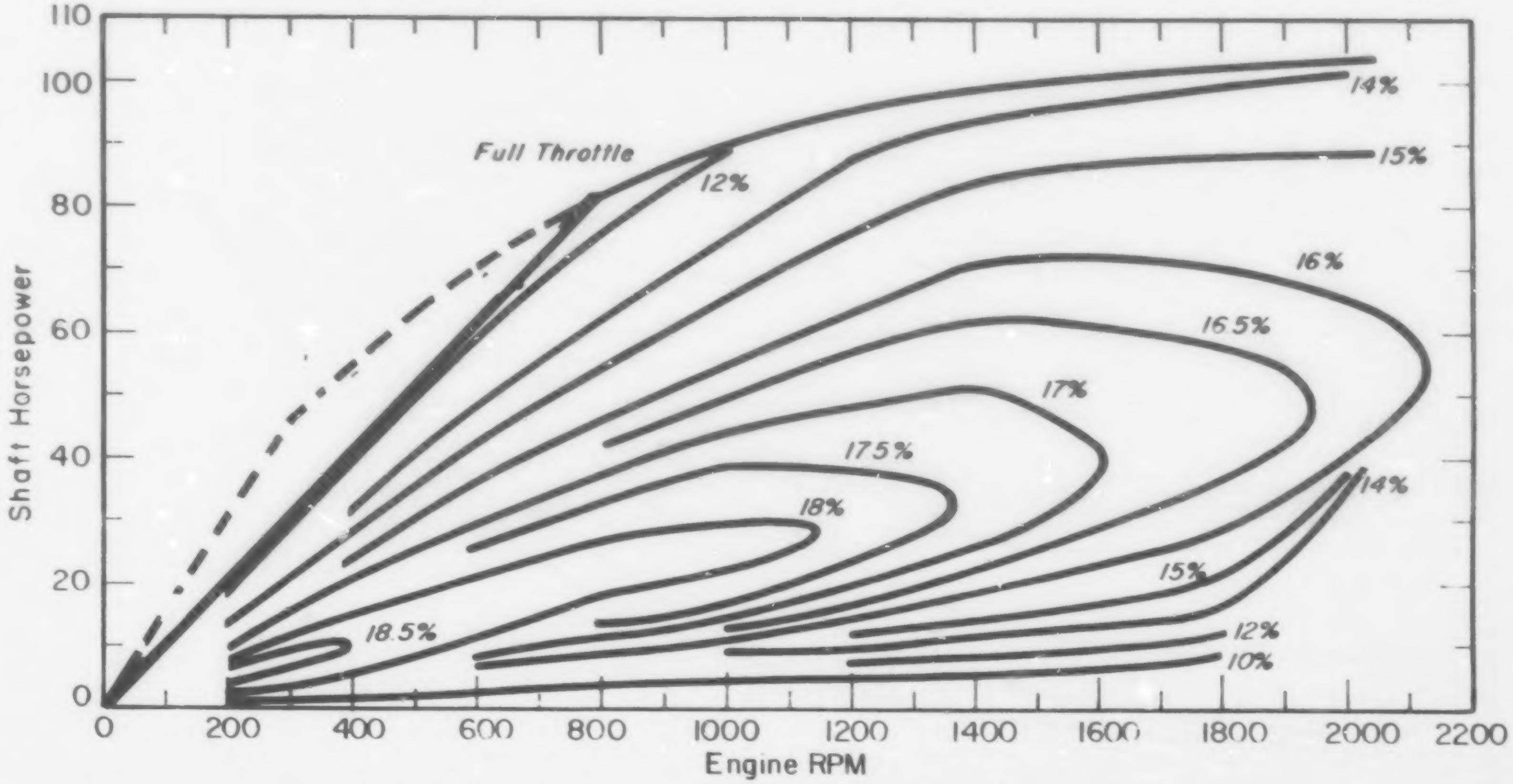


Figure 4.1.2 Performance Map with 184 CID Engine and Maximum Intake Ratio of 0.29.

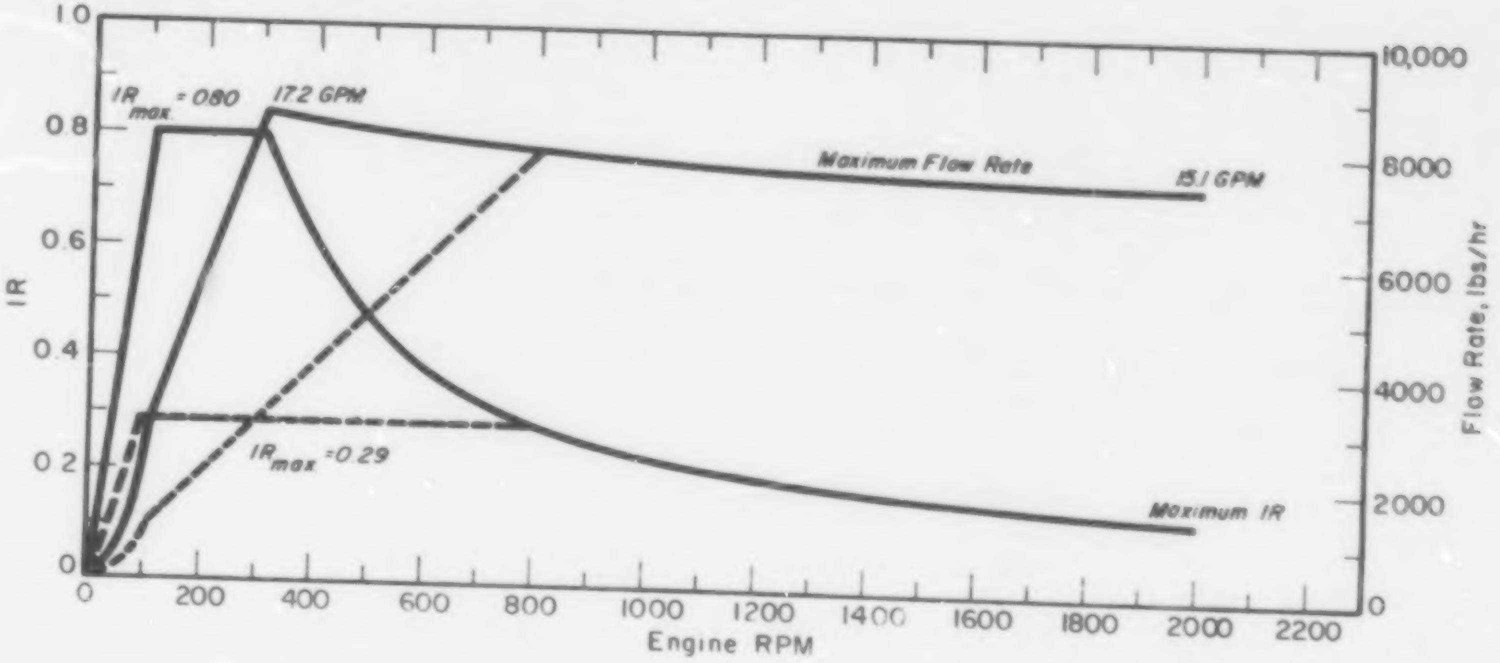


Figure 4. 1. 3 Maximum Intake Ratio and Maximum Organic Flow Rate as Functions of Engine Speed.

TABLE 4.1.1

SYSTEM PERFORMANCE COMPUTER PROGRAM PRINTOUT FOR DESIGN POINT CONDITIONS

```
***** INTAKE RATIØ =  .13687      PISTØN SPEED =  1000.0      *****

THETA =  43.427

P1 =  500.00      T1 =  550.00      H1 =  123.40      S1 =  .31553E-01  V1 =  .18730

P2 =  58.811      T2 =  375.80      H2 =  84.024      S2 =  .31553E-01  V2 =  1.7332

PCØND =  24.996      TCOND =  216.20      HVAP =  39.633

TSUB =  196.20      VSUB =  .16444E-01  HSUB = -125.84

H3 =  77.182      H4 =  43.388      H8 = -124.03      H9 = -90.236

QRIV =  1.6132      QREV =  .45542      Q1 =  5.3661      Q2 =  2.4396

FLØWRATE =  7377.4                WSHAFT =  37.399              WES =  48.287

WNET =  35.584                    H.P. =  103.15

EFFTH =  .84603      EFFME =  .91546      EFFALL =  .77451

WP =  1.8142      EFFPM =  .79691

QREG =  33.794      EFFREG =  .90000

QBØILER =  213.63611            HBØILER =  1576069.0       EFFBØILER =  .82500

QCØND =  169.23                 QCØNDTØT =  1248484.0

IMEP =  127.35      RPM =  2000.0

**** CYCLE EFFICIENCY =  .16657      *****

***** ØVERALL EFFICIENCY =  .13742      *****
*STØP* 0
```

TABLE 4.1.2

DESIGN POINT SPECIFICATIONS

Working Fluid	Thiophene
Boiler Outlet Temperature	550°F
Boiler Outlet Pressure	500 psia
Boiler Heat Transfer Rate	1.58×10^6 Btu/hr
Boiler Efficiency (HHV)	82.5%
Engine Displacement	184 in^3
Engine Speed	2000 rpm
Engine Piston Speed	1000 ft/min
Engine Horsepower Less Feedpump Power	103.2 hp
Engine Thermal Efficiency	84.6%
Engine Mechanical Efficiency	91.5%
Engine Overall Efficiency	77.5%
Engine IMEP	127.4 psi
Regenerator Effectiveness	90.0%
Regenerator Heat Transfer Rate	0.249×10^6 Btu/hr
Condensing Temperature	216.2°F
Condensing Pressure	25.0 psia
Subcooled Liquid Temperature	196.2°F
Condenser Heat Transfer Rate	1.25×10^6 Btu/hr
Organic Mass Flow Rate	7377 pounds/hr
Organic Volumetric Flow Rate	15.1 gallons/min
Feedpump Overall Efficiency	79.7%
Feedpump Power	5.25 hp
Cycle Efficiency	16.7%
Overall Efficiency	13.7%

4.2 SAFETY CONSIDERATIONS FOR THIOPHENE WORKING FLUID

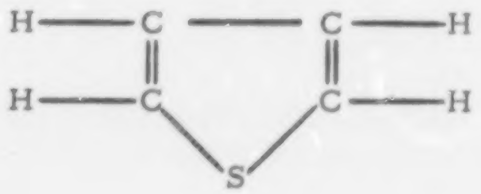
4.2.1 Introduction

As mentioned previously, greater emphasis was placed on the use of a working fluid which provides a minimum cost system with excellent performance and efficiency characteristics, than on minimum hazard. At the same time, it was not desirable to select a fluid which was obviously unacceptable from a hazard point of view, such as a highly poisonous material (where slight leakage of the working fluid would represent a lethal threat to those around the car). The opposite extreme of requiring complete non-toxicity and non-flammability is equally undesirable, since this requirement restricts consideration to one, and only one, working fluid, steam; the impracticality of using steam for portable power systems has been amply demonstrated. Thiophene was, therefore, selected as the state-of-the-art working fluid with characteristics fulfilling the development goals, and with acceptable flammability and toxicity, at least for the initial demonstration prototypes. This selection has been justified by two years of continuous use of thiophene at Thermo Electron Corporation in Rankine-cycle component and system development without accident or injury to any personnel and without special safety precautions except to insure that the development laboratories are adequately ventilated. This work has involved operation of test loops and disassembly and inspection of test engines.

4.2.2 Flammability and Toxicity of Thiophene

Relatively little work has been performed on the quantitative flammability and toxicity characteristics of thiophene. In this section, a summary of the available information will be presented.

Thiophene has the chemical formula:



In Table 4.2.1, a summary of its flammability characteristics and a comparison with common liquid fuels are presented. Definitions of the flammability parameters used in evaluating the flammability hazard of fluids are summarized in Appendix A. It is apparent that thiophene lies between gasoline and kerosene as a fire hazard, using flash point as the most appropriate criterion for hazard comparison. While a higher flash point would be advantageous with respect to low temperature leaks, it is doubtful if any organic working fluid would have a flash point above the maximum cycle temperature. Since the working fluid could be released as a vapor at the maximum cycle temperature in case of a structural failure in the engine or in the tubing connecting the boiler to the engine, such as might occur if the vehicle were involved in an accident, the fire hazard in the engine compartment with a Rankine-cycle system using organic working fluid has to be considered as higher than in a conventional internal combustion engine burning gasoline. In Section 4.2.3, system design features to alleviate this fire hazard are discussed. It should be noted that a considerably less flammable fuel can be used for the Rankine-cycle system than for gasoline-fueled internal combustion engines.

TABLE 4.2.1

FLAMMABILITY CHARACTERISTICS OF THIOPHENE AND COMMON FUELS

Characteristic	Thiophene	Gasoline	Kerosene (No. 1 Fuel Oil)	Diesel Fuel (No. 2 Fuel Oil)
Heat of Combustion (HHV), Btu/lb	14350	20460	18500	20500
Flash Point, CC, °F	20°F(2)	-50°F(1)	100-165(1)	100-190(1)
Fire Point, °F	20°F(2)	–	–	–
Autoignition Temp., °F	755(2)	495(1)	490(1)	494(1)
Explosive Range	–	1.3-6.0%(1)	1.16-6.0%(1)	–

Considerable work, though by no means complete, has been carried out on the toxicity of thiophene. A review of this work is given in Table 4.2.2; in Appendix B, an independent summary of the thiophene toxicity is given and compared with that of gasoline. It appears that the only significant problem area with thiophene toxicity is vapor inhalation; a concentration of 2,900 ppm caused severe effects so that an acceptable concentration would be considerably less than this level. No information is available on the chronic effects of repeated or prolonged exposure to low concentrations (100 to 1,000 ppm) of thiophene. It would appear that thiophene is somewhat more toxic with respect to vapor inhalation than gasoline, but not by a large factor.

To provide a perspective with respect to the toxicity hazard, the internal combustion engine in well-tuned 1970 automobiles emits about 10,000 ppm of CO. In Figure 4.2.1, a plot is presented of the effects of different CO concentrations on humans. Sax[1] describes an hour's exposure to 1,000 - 1,200 ppm as dangerous, while exposures to 4,000 ppm are fatal to humans in less than an hour. It is apparent that the CO concentration in the exhaust from well-tuned internal-combustion powered automobiles is much greater than that required for death to humans, even on short-term exposure. It is also apparent that CO is considerably more toxic than thiophene, based on the available data. Thus, current automobiles, in operation, continuously emit a poison in much greater concentrations than are required for human death. Fatalities due to accidental CO exposure from automobile exhausts are relatively rare, though they do occur.

TABLE 4.2.2

SUMMARY OF TOXICITY OF THIOPHENE

Source of Information	Oral Toxicity	Skin Absorption	Skin Irritation	Eye Irritation	Vapor Inhalation	Subcutaneous Injections
Report by Younger Laboratories on work performed for Monsanto Company, St. Louis, Missouri, 1965.(2)	LD_{50} = 3.1 gms/kg of body mass for rats; lower and upper limits = 2.5 gms/kg and 3.8 gms/kg, compound classed as slightly toxic. LD_{50} = dose required for 50% fatalities in test animals.	3.2 $\frac{gms}{kg}$ < Minimum Lethal Dose for rabbits < 5.0 $\frac{gms}{kg}$ Compound classed as slightly toxic.	Compound classed as moderate skin irritant for rabbits. Average maximum score was 3.6 out of possible 8 in 24 hours.	Compound classed as moderate eye irritant for rabbits; average maximum score was 37.0 out of possible 110 in 24 hours.	Concentration of 91,000 ppm resulted in death of rats in 20-25 minutes. This is not unexpected in view of the very high concentration of vapor (9.4% by volume) produced in the test. Other volatile materials considered relatively low in toxicity, e.g., gasoline and cyclohexane, are reported to have acute lethal effects in concentrations at or near 5% by volume.	—
Handbook of Laboratory Safety(3) "Entry based upon analogy with a closely similar structure, or on other estimate believed sound."	Minor residual injury may result from some accidental exposures if no treatment is applied.	Minor residual injury may result from some accidental exposures if no treatment is applied.	No residual injury is to be expected from accidental exposure even if no treatment is applied.	Minor residual injury may result from some accidental exposures if no treatment is applied.	Minor residual injury may result in spite of prompt treatment.	—
F. Flury and F. Zernick(4)	—	—	—	—	2,900 ppm caused loss of consciousness and, under certain circumstances, death of mice. 8,700 ppm caused death of all mice in 20 to 80 minutes.	—
A. [illegible]	—	—	—	—	—	Dog weight - [illegible] kg, one injection of approximately 2 gms per day. First injection - no effect. Second injection - loss of desire for eating. Third injection - lack of control and paralysis, dogs could not walk; no desire for food. After doses over 10 gms, the dogs die of paralysis.
The Merck Index 8th Edition						Minimum lethal dose in rabbits - 0.84 gms/kg.

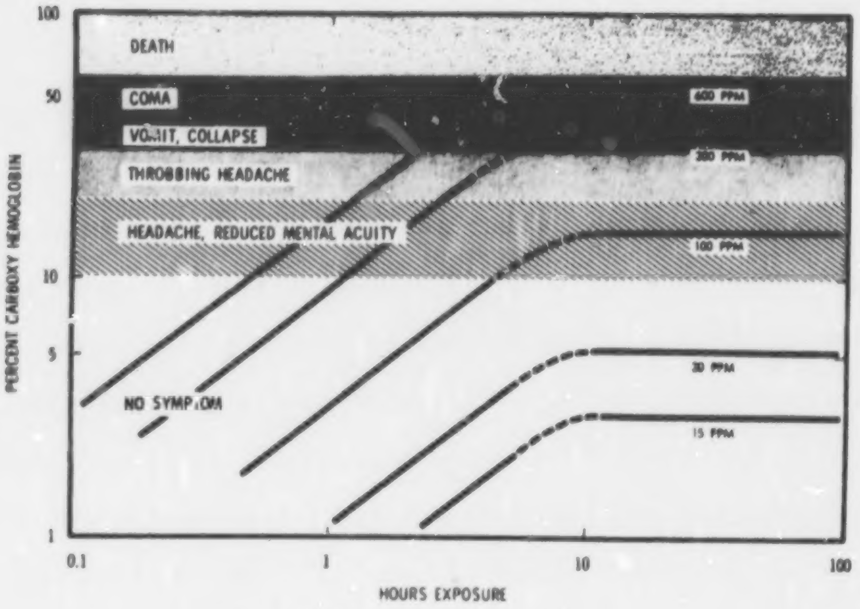


Figure 4.2.1 Effect of CO Concentration and Time on Carboxy Hemoglobin.[7]

4.2.3 System Design Concepts to Minimize Hazard from Flammable and Toxic Working Fluid

From the preceding discussion, it is apparent that thiophene does represent a flammability and toxicity hazard. In this section, a discussion is presented of system design concepts to minimize this potential hazard.

4.2.3.1 Thiophene Leakage from the System

The system is designed as a completely sealed, leak-tight system, and must be maintained basically vacuum-tight for satisfactory operation of the system. Only one rotary shaft seal is used; to minimize the potential for leakage from this source, a double-face seal is used with pressurized buffer fluid of the system lubricant between the seals. This oil pressure is maintained above the working fluid pressure at all times so that the small leakage normally encountered with a face seal results in lubricant entering the system rather than thiophene leaving the system. All static seals will be the equivalent of an "O" ring vacuum tight seal using a material (Viton) compatible with thiophene and having a long "shelf-life" (resistant to oxidation and other environmental conditions). It should also be noted that the system is at positive gauge pressure only when operating. When shut down and cooled, the thiophene pressure is much less than atmospheric pressure so that any leakage is either air-leakage or oil-leakage into the system rather than thiophene-leakage out of the system.

The entire system is conservatively designed from stress considerations to prohibit rupture of the various components. Part of the manufacturing procedure of the system should be pressure testing of all components according to ASME code specifications and complete vacuum leak-testing of the complete assembled system, followed by hot operation of the system for 15 minutes. Surface coating can be used on carbon steel surfaces to prevent corrosion over the 10 year life of an automobile.

In case of accident, the most vulnerable component is the condenser, which is placed in the same position as the radiator on an internal combustion engine-powered automobile. If this placement proves unacceptable, an alternative is to use water as a heat-transfer intermediate, as illustrated in Figure 4.2.2, so that the thiophene condenser placement can be in the rear of the engine compartment. This option, of course, results in a significant increase in the cost of the system and should not be considered unless experimental evidence indicates that front end placement of the thiophene condenser would result in an unacceptable hazard in case of front end collisions.

An additional factor of importance is to minimize the thiophene inventory in the system. The fluid inventory in the design presented in this report for the 100 shaft horsepower system is 30 lbs. It is expected that this inventory can be reduced by 5 to 10 lbs with refinement in the system design. It is unlikely that the fluid inventory can be reduced below about 20 lbs.

Safety controls must be incorporated to prevent system damage in event of malfunction of the normal control system. For critical parameters, such as boiler pressure and condenser pressure, two levels of safety control are proposed: one, to detect off-design performance and shut the system down before a serious operating condition is reached; the other, final pressure relief to prevent system rupture and to control the position of thiophene release from the system, minimizing the potential for ignition of the thiophene. For example, a rupture disc is used on the condenser to prevent condenser rupture in the event of simultaneous failure of the normal control system and of the safety control to shut down the system. The exhaust rupture disc would be ducted beneath the car to reduce the potential for ignition of the thiophene by electric sparking or

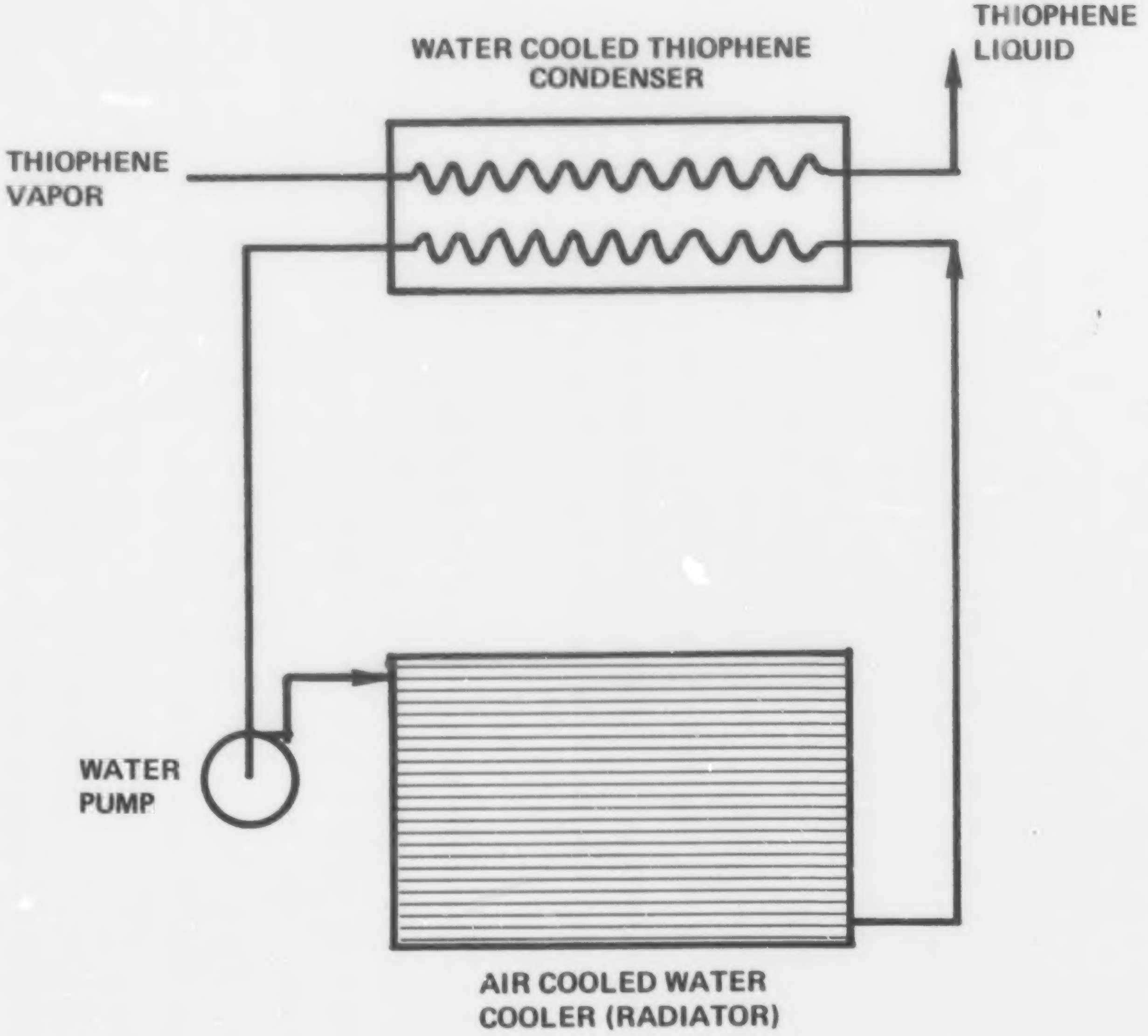

WATER COOLED THIOPHENE CONDENSER
THIOPHENE LIQUID
THIOPHENE VAPOR
WATER PUMP
AIR COOLED WATER COOLER (RADIATOR)

Figure 4.2.2 Alternate Thiophene Condenser to Reduce Front-End Collision Hazard.

other potential sources of ignition in the engine compartment. The high density of thiophene vapor and the cooling air movement through the engine compartment would prevent the thiophene vapor from re-entering the engine compartment in appreciable concentration.

4.2.3.2 Effects of Leakage

Even with conservative design, leaks can develop in the system, from either normal use or accidents which result in rupture of parts of the sealed system. It should be noted that the system, when cold, is at subatmospheric pressure so that the thiophene will not leak from the system under these conditions.

To aid in detection of small leaks in the system, a small concentration (ppm) of an odiferous material can be added to the highly purified thiophene (which has a relatively faint odor). This procedure is identical to that used for natural gas supplied for residential use, where highly odiferous mercaptans are added to the natural gas for detection of leaks before dangerous concentrations are encountered. This approach, coupled with the subatmospheric pressure of the cold system, should greatly reduce the hazard when the vehicle is parked in a confined space such as a garage.

The development of small leaks when the vehicle is operating should also be a minor hazard, since the large volume of condenser cooling air will sweep the thiophene from the engine compartment with large dilution of the thiophene vapor in the air.

The principal hazard thus comes from large leaks resulting from rupture of lines, with the system operating, where flammable thiophene-air mixtures can be formed and toxic concentrations of thiophene vapor can exist around the vehicle. No easy solution to this situation exists. To reduce flammability hazards, the engine compartment could be equipped with a CO_2 system which would be released by a sensor located in the air duct leading to the burner. The sensor could be a heated, catalytic coil located in the engine compartment which would burn any flammable thiophene-air mixture which came into contact with it, thus raising the temperature in the sensing circuit and simultaneously releasing the extinguisher and shutting the burner down. An inertia switch could also be incorporated for immediate release of the fire extinguisher system in event of accident. Considerable experimentation will be required to establish the practicality of this approach.

The toxicity hazard in event of a large leak is difficult to define. Once the vapor leaves the engine compartment, dilution with air can be expected, probably reducing concentrations to tolerable levels around the vehicle. In the passenger compartment, except in a convertible, an additional dilution would be expected relative to that outside the car. Experimental work is required to determine the actual concentrations. The odor of the thiophene, or of odiferous additives, will provide positive indication of the presence of thiophene vapor, in any case, so that the vehicle occupants can move away from the vehicle to a point where the vapor concentration is very low.

The optimum solution to the hazard problem is to find a working fluid which provides the thermodynamic and economic advantages of thiophene, and which is also fire-resistant and much less toxic. Such a fluid does not now exist, however, and a major effort is required for the development of such a fluid. Discussion with the various chemical companies involved in thermodynamic fluid development indicates a high probability for synthesis of such a fluid once the practicality of the Rankine-cycle system from performance and economic considerations has been established and the incentive for the large investment required for development of such a fluid can be justified. With development of such a fluid, the automobile with a Rankine-cycle propulsion system should be safer than the present internal combustion engine, due to elimination of the CO toxicity hazard and the use of a fuel considerably less flammable than gasoline. Even with the thiophene working fluid, the overall hazard of the Rankine-cycle system may be comparable to the internal combustion engine system, based on the reduction in deaths due to CO poisoning, which should not occur at all for the Rankine-cycle system, and to gasoline fires in accidents, which should be significantly reduced due to use of a less flammable fuel.

4.3 ENGINE (EXPANDER) DESIGN

4.3.1 Performance Estimates

In order to determine proper operating speeds and inlet and exhaust valve sizes, a simple analytical model of the expander was constructed, and mechanical and indicated efficiencies were predicted for a range of speeds and loads.

a. Prediction of Mechanical Efficiency

The mechanical efficiency is defined as follows:

$$\eta_m = \frac{\text{Brake Mean Effective Pressure (BMEP)}}{\text{Indicated Mean Effective Pressure (IMEP)}}$$

Data from Thermo Electron Rankine expanders and internal combustion engine data were used to derive the following expression for mechanical efficiency as a function of mean piston speed [S = (2) (stroke) (rpm)] and IMEP:

$$\eta_m = 1 - S\left[\frac{174}{\text{IMEP}} + 1.6\right] \cdot 10^{-5} + \frac{5.46}{\text{IMEP}} + 0.012.$$

This expression is plotted for a range of S and IMEP in Figure 4.3.1.

b. Prediction of Indicated Efficiency

The indicated efficiency of the expander is defined as follows:

$$\eta_i = \frac{\text{IMEP}}{\text{Isentropic Indicated Work}}$$

The analysis is based on the following assumptions:

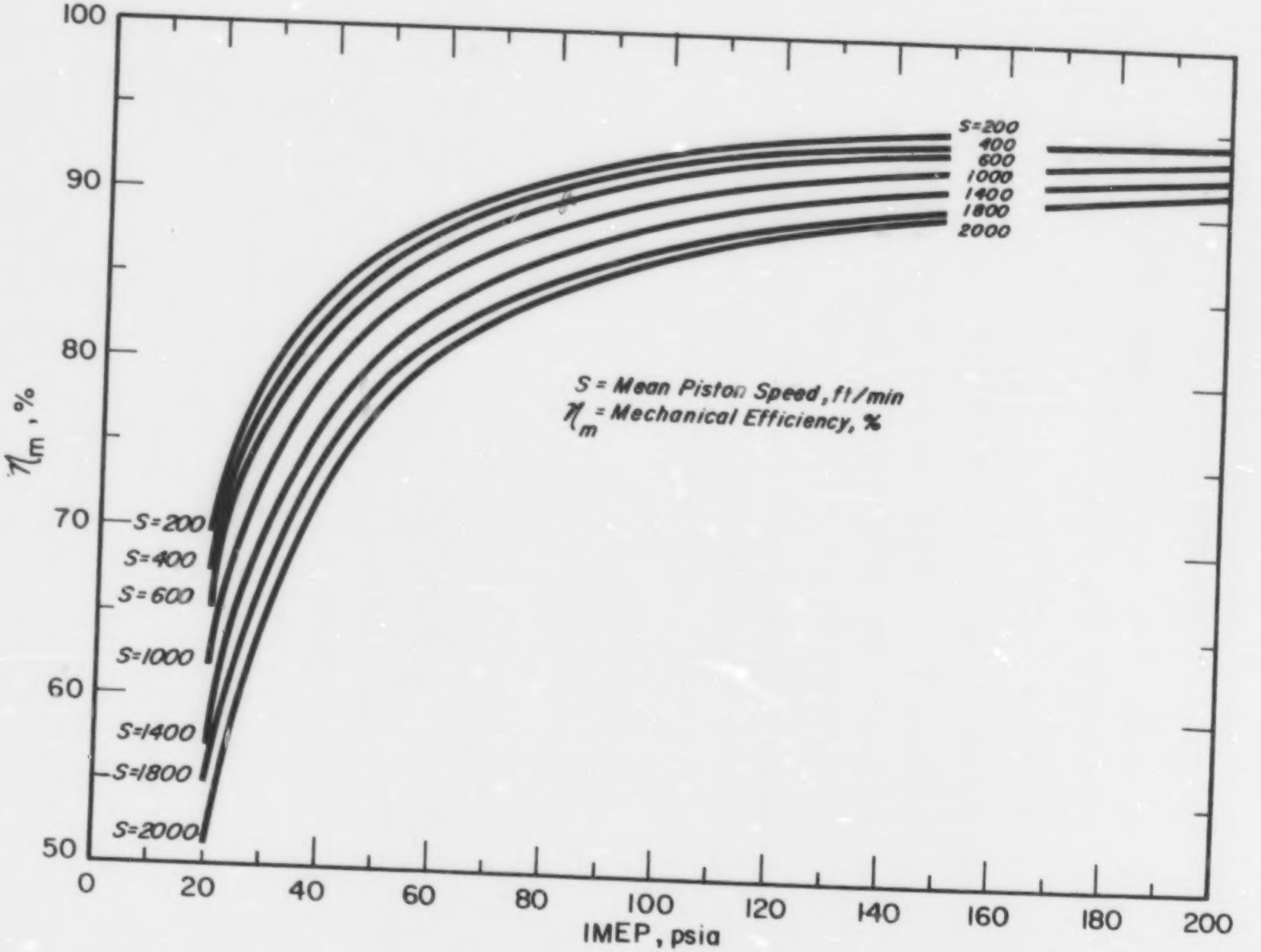


Figure 4. 3. 1 Engine Mechanical Efficiency Variation with IMEP for Various Piston Speeds

(1) There are three types of losses in the expander: throttling through the inlet valve, throttling through the exhaust valve, and heat losses.

(2) No blowdown losses due to exhaust ports being uncovered before bottom center occurs.

(3) No recompression occurs.

(4) The expander has zero clearance volume.

(5) The pressure losses are assumed to be such that they appear as straight lines on the P-V diagram, as shown in Figure 4.3.2.

Assuming incompressible flow through the inlet valve, the orifice equation can be used to derive the following expression for the average pressure loss through the inlet valve:

$$\Delta P_{iv} = \left(\frac{A_p}{A_{iv}}\right)^2 \frac{1}{8g} \left(\frac{S}{C}\right)^2 \frac{v_i}{v_2^2} \left(\frac{360}{\theta}\right)^2 ,$$

where

θ = $\cos^{-1} (1 - 2\, v_i/v_2)$

A_p/A_{iv} = ratio of piston area to average inlet valve area (see Figure 4.3.3 for explanation)

g = acceleration due to gravity

S = mean piston speed

C = inlet valve flow coefficient (0.6 assumed)

v_i = inlet specific volume

v_2 = release specific volume

θ = crank angle for intake opening

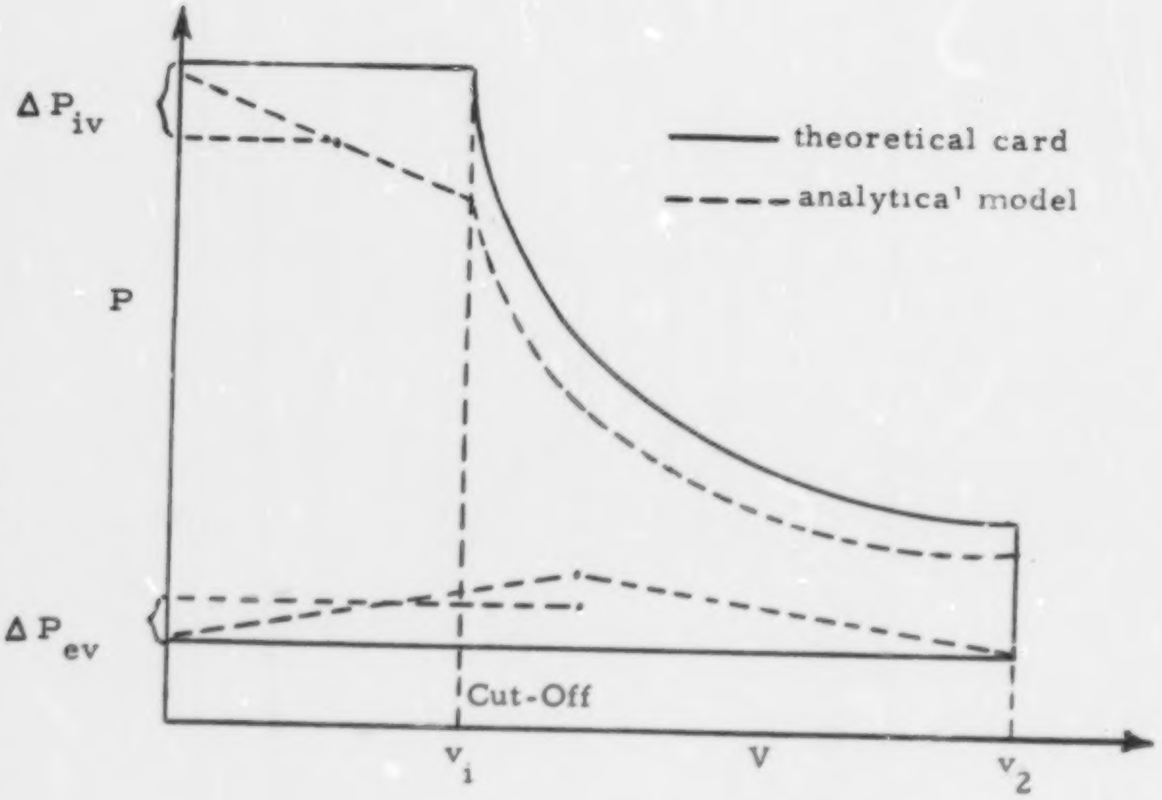

ΔP_iv
P
ΔP_ev
theoretical card
analytical model
Cut-Off
v_1
V
v_2

Figure 4.3.2 Sketch Illustrating Analytical Model Used for Expander.

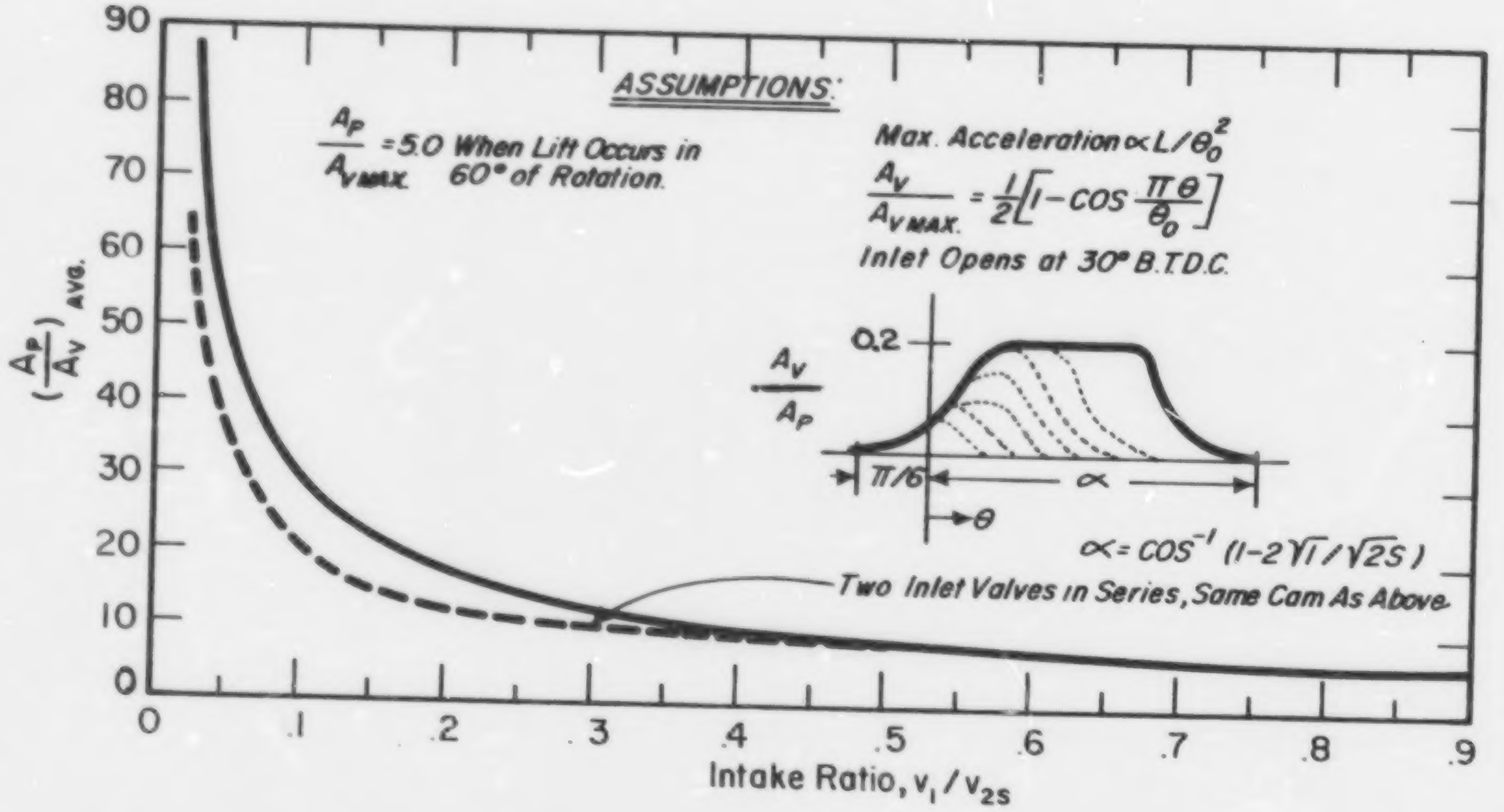


Figure 4.3.3 Variation of Piston Area to Valve Area Ratio with Intake Ratio of Engine

The work lost during the intake process is:

$$\Delta W_{iv} = (\Delta P_{iv})(v_i) .$$

The exhaust process can be treated the same way, and the resulting relationship is:

$$\Delta P_{ex} = \frac{1}{2g}\left(\frac{A_p}{A_{ev2}}\right)^2 \left(\frac{1}{C_2}\right)^2 \frac{S^2}{v_2} - \frac{1}{g}\left(\frac{EO}{360}\right)\left(\frac{A_p A_{evl}}{A^2_{ev2}}\right)\left(\frac{C_1}{C_2^2}\right) S\sqrt{\frac{2g\ \Delta P}{v_2\ 2}} ,$$

where

A_{ev2} = area of auxiliary exhaust ports

A_{evl} = area of blowdown exhaust port

C_1 = flow coefficient of blowdown exhaust ports (0.6)

EO = crank angle through which blowdown exhaust ports are uncovered (80°)

ΔP = release pressure - condenser pressure.

The work lost during exhaust is

$$\Delta W_{ex} = (\Delta P_{ex})(v_2).$$

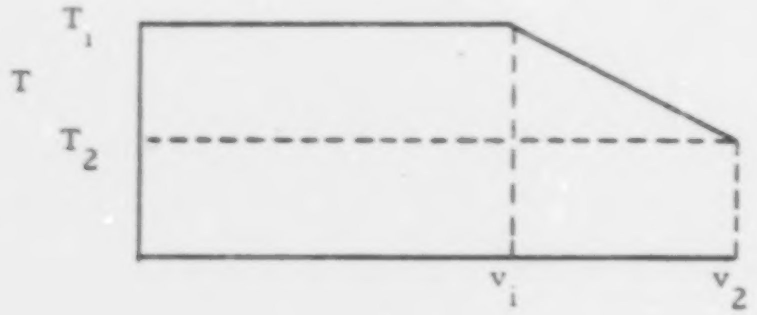
The heat loss correlation was assumed to be of the usual form of

$$Nu = \text{Constant} \times Re^n .$$

The bore was taken as the characteristic dimension in the Nusselt and Reynolds numbers, and the constant was determined by fitting Thermo Electron data. The exponent, n, was taken as 0.75, a value used in internal combustion engines. The above relationship has a heat transfer coefficient based on piston area; thus,

$$Q = hA_p (T_G - T_w) .$$

The mean gas temperature was taken as the arithmetic mean of the inlet temperature and temperature at beginning of blowdown assuming a straight line temperature drop during expansion:



$$T_G = T_i (v_i/v_2) + \frac{(v_2 - v_i)}{2 v_2} (T_i + T_2)$$

The mean wall temperature is based on test data taken from a single cylinder engine operating at the same inlet conditions with a number of thermocouples along the cylinder The resulting expression is a function of intake ratio only, over a fairly small range:

$$T_w = 400 + 156 (v_i/v_2)^2$$

The heat loss in Btu/cycle for a 4 0 inch bore is:

$$q = 1.48 \quad \frac{kv_2}{S}\left(\frac{S}{v_2 \mu}\right)^{0\ 75} \qquad (T_G - T_w) ,$$

where

k = vapor conductivity at T_G

μ = vapor viscosity at T_G .

With the above three expressions, the indicated efficiency can be calculated for any piston speed and intake ratio. The process is iterative, in that v_2 and P_2 are functions of the pressure loss through the inlet valve. The above expressions were incorporated into a computer program for calculating the overall cycle and engine efficiencies. The overall engine efficiency as a function of piston speed, at a fixed intake ratio corresponding to an IMEP ≈ 125 psi, is shown in Figure 4.3.4.

4.3.2 Engine Configuration

As a result of the rapid decline in overall engine efficiency with piston speed above 1000 ft/min (see Figure 4.3.4), this value was taken as the piston speed at the engine design condition of 103.2 bhp at 95 mph. The IMEP and BMEP are determined by the cycle design condition (piston speed and intake ratio); the BMEP and piston speed determine the piston area required to develop the desired horsepower. A 90° V of four cylinders was chosen as being reasonably compact without a large number of moving parts. With four cylinders, the resulting bore is 4.42 inches. The mean piston speed and engine speed are related through the following expression:

$$S = 2\ LN,$$

where S = mean piston speed,

L = stroke,

N = rpm.

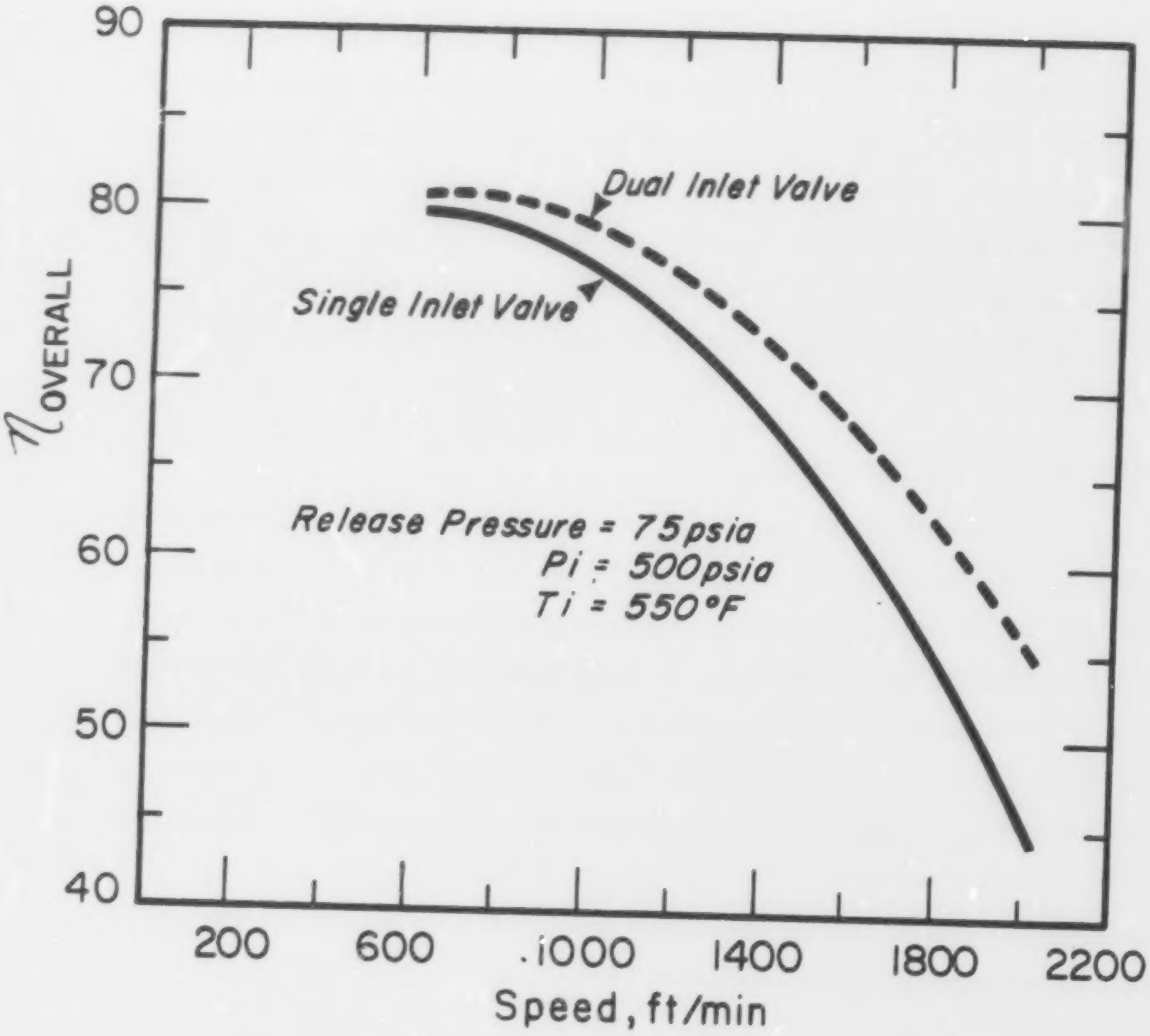


Figure 4. 3. 4 Overall Engine Efficiency Variation with Piston Speed

A design point speed of 2000 rpm was chosen as being a reasonable maximum, considering the problem of driving the inlet valves (see following section). This speed also resulted in a reasonable bore/stroke ratio, giving a stroke of 3.0 inches.

The resulting expander dimensions and design point operating conditions are given as follows:

Configuration	Four cylinders, 90°V
Bore	4.42 inches
Stroke	3.0 inches
Displacement	184 in^3
BHP (design)	103 at 2000 rpm
IMEP (design)	127 psi
BMEP (design)	117 psi

4.3.3 Expander Intake Valving

The discussion of Section 5 has indicated the importance of variable intake valve closing (or cut-off). The size of the intake valve is determined from Figure 4.3.3 once the piston area and maximum piston speed are known. A valve providing an average opening area smaller than that indicated in Figure 4.3.3 would give lower expander efficiencies than were assumed in the design point calculations.

a. Comparison with Internal Combustion Engine Valving

The resulting inlet valve must be about 1.25 inches in diameter and have a lift of 0.30 inches. This valve is, therefore, comparable in size (and weight) to the valve of an internal combustion engine of the same bore. The stress level in a cam-operated valve is proportional to the lift divided by the valve event squared. The cam event in typical four-stroke internal combustion engines is on the order of 120°, whereas in the Rankine-cycle expander the design point intake ratio of 13.7% implies a valve event of no more than 60 or 70°. One other problem associated with expander valving which does not occur in the internal combustion engine is the high pressure against which the valve must open (or close, depending on whether the valve opens inward into the cylinder or outward into the inlet port). This has led to the development of two types of pressure-balanced valves, shown in Figure 4.3.5.

b. Mechanically-Operated Valving

Two types of mechanically-operated systems were considered in detail. The first was a three-dimensional cam system in which the timing would be varied by sliding the cam shaft along its axis. Analysis showed that such a system would not be feasible even if a three-dimensional cam were economically practical, since the cam base circle diameter would have to approach the size of the bore of the engine itself to give reasonable loading on the cam.

The second, more promising approach consists of operating two concentric inlet valves in a series arrangement, as shown schematically in Figure 4.3.6. The number 1 valve in Figure 4.3.6 is driven by camshaft 1, which has a fixed angular relationship with the crankshaft; the number 2 valve is driven by camshaft 2, which has a variable angular relationship with the crankshaft. The total valve event is then controlled by the opening of valve 1 and the closing of valve 2, and the cut-off point is determined by the overlap of the two valves. With this system, long cam events are possible, resulting in much lower stresses in cam and valve gear and larger flow areas (at low cut-off) than could be accomplished with a single valve (see Figure 4.3.3), although this advantage would probably be partially negated due to a lower effective flow coefficient for the two valves in series.

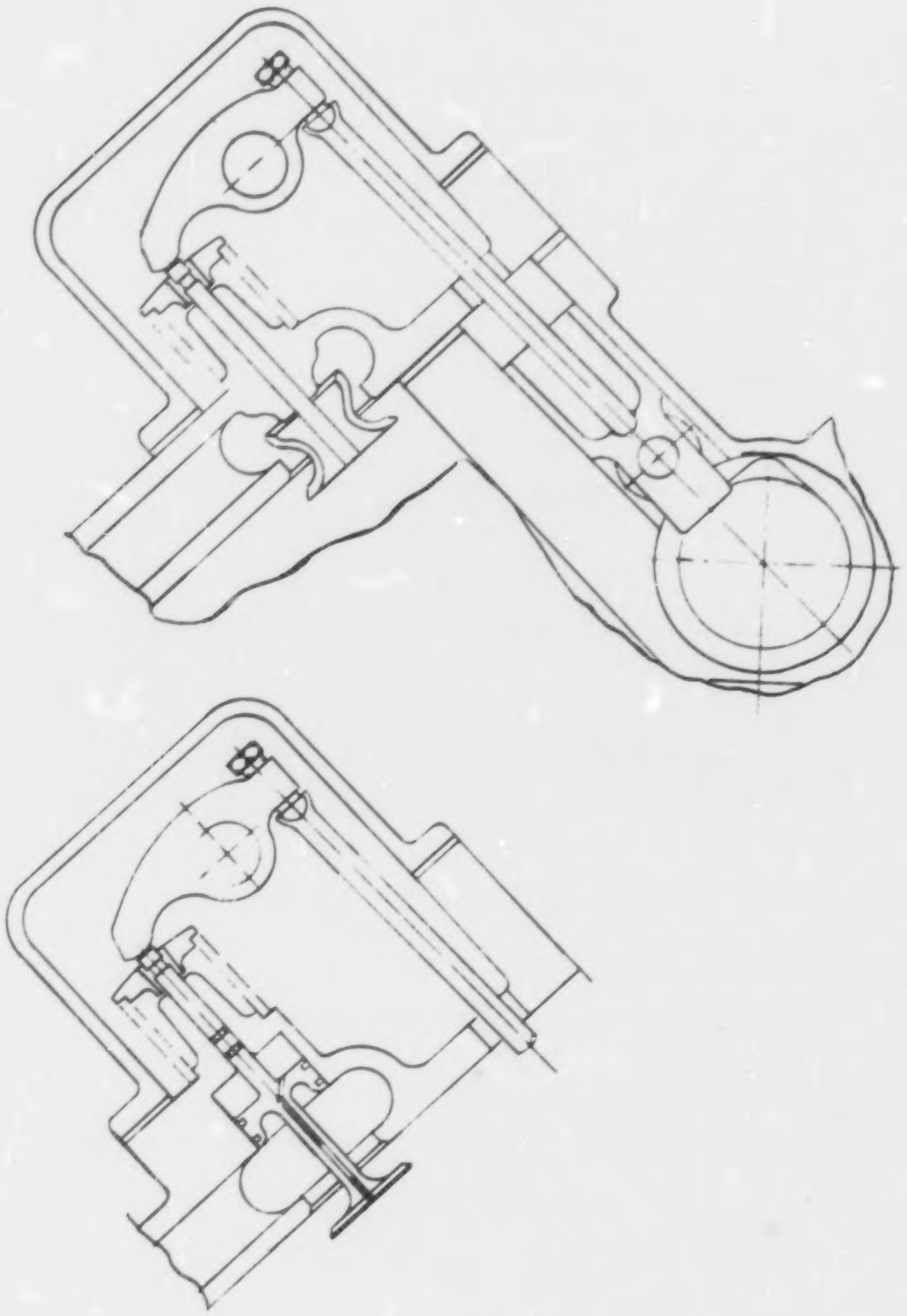


Figure 4.3.5 Alternative Balanced Inlet Valves.

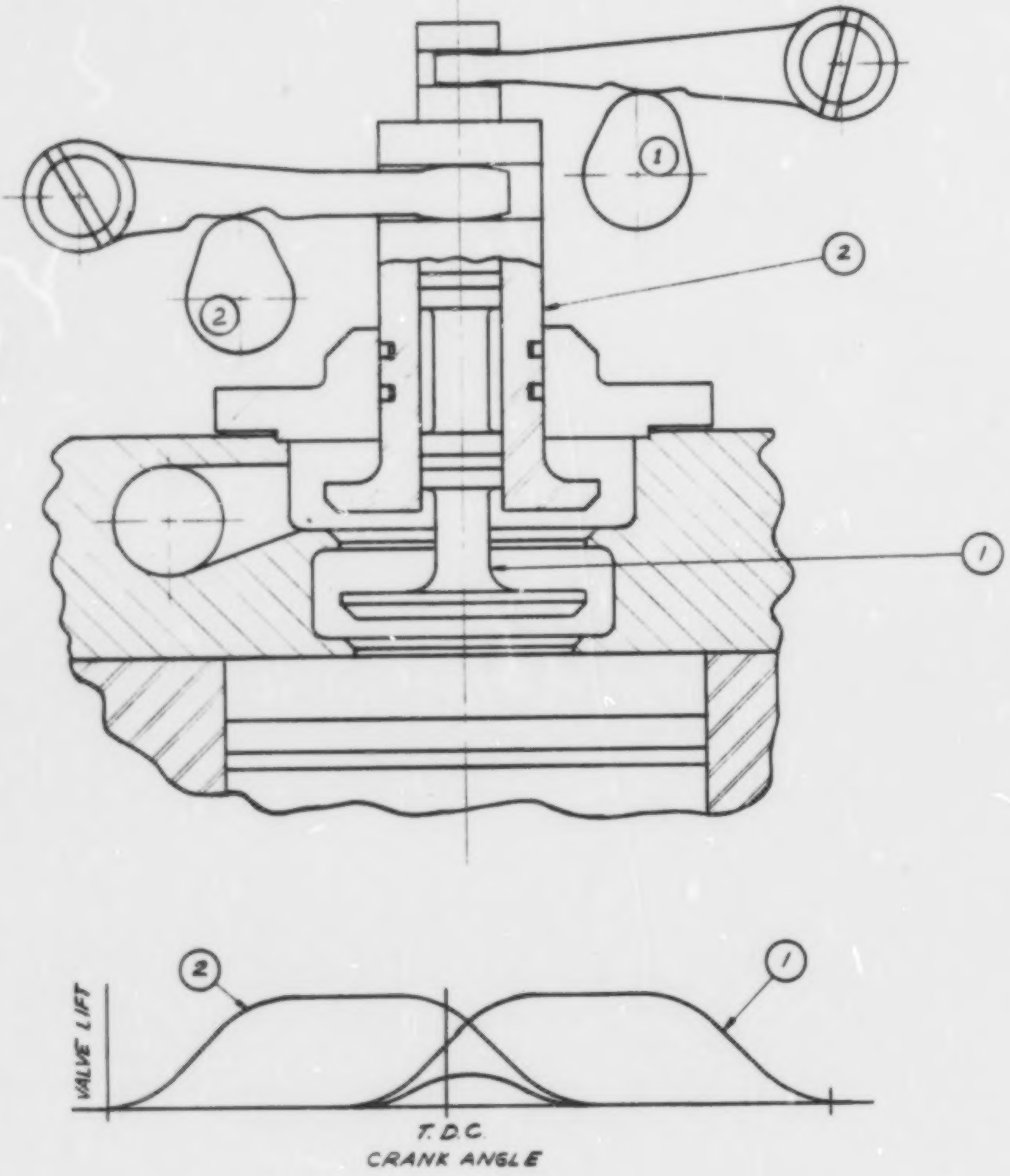

1
2
2
1
2
1
VALVE LIFT
T.D.C.
CRANK ANGLE

Figure 4.3.6 Two Inlet Valves, In Series

A design study of the two valves in series approach was carried out on an in-line four cylinder expander with the same bore and stroke as the V expander. This design is shown in Figure 4. 3. 7. Because of the very large springs required to overcome inertia and pressure forces (these valves cannot easily be pressure-balanced), a two-cam (or desmodromic) approach was adopted, with one cam opening the valve and the second closing it. Both cams are on the same shaft and actuate a single rocker. Using a cam rather than a spring to close the valve has the added advantage of giving more rapid closing and sharper cut-off.

c. Hydraulically Operated Valving

(1) Directly Actuated

The approach illustrated in Figure 4. 3. 8 consists of a cam-operated plunger pump which operates on a hydraulic column to actuate the inlet valve. The angular orientation of the plunger in its bore determines its effective stroke and hence the intake valve event. This system is similar in many ways to diesel engine injection equipment, and a manufacturer of such equipment feels that this system is feasible. The peak pressure in the hydraulic column would be on the order of 5000 psi, and a spring force of about 400 pounds would be required to accelerate the intake valve during closing.

(2) Pilot Operated

This system is shown schematically in Figure 4. 3. 9. A single rotary valve (two are shown in the figure for clarity) delivers high pressure oil from a gear pump and accumulator to alternate sides of

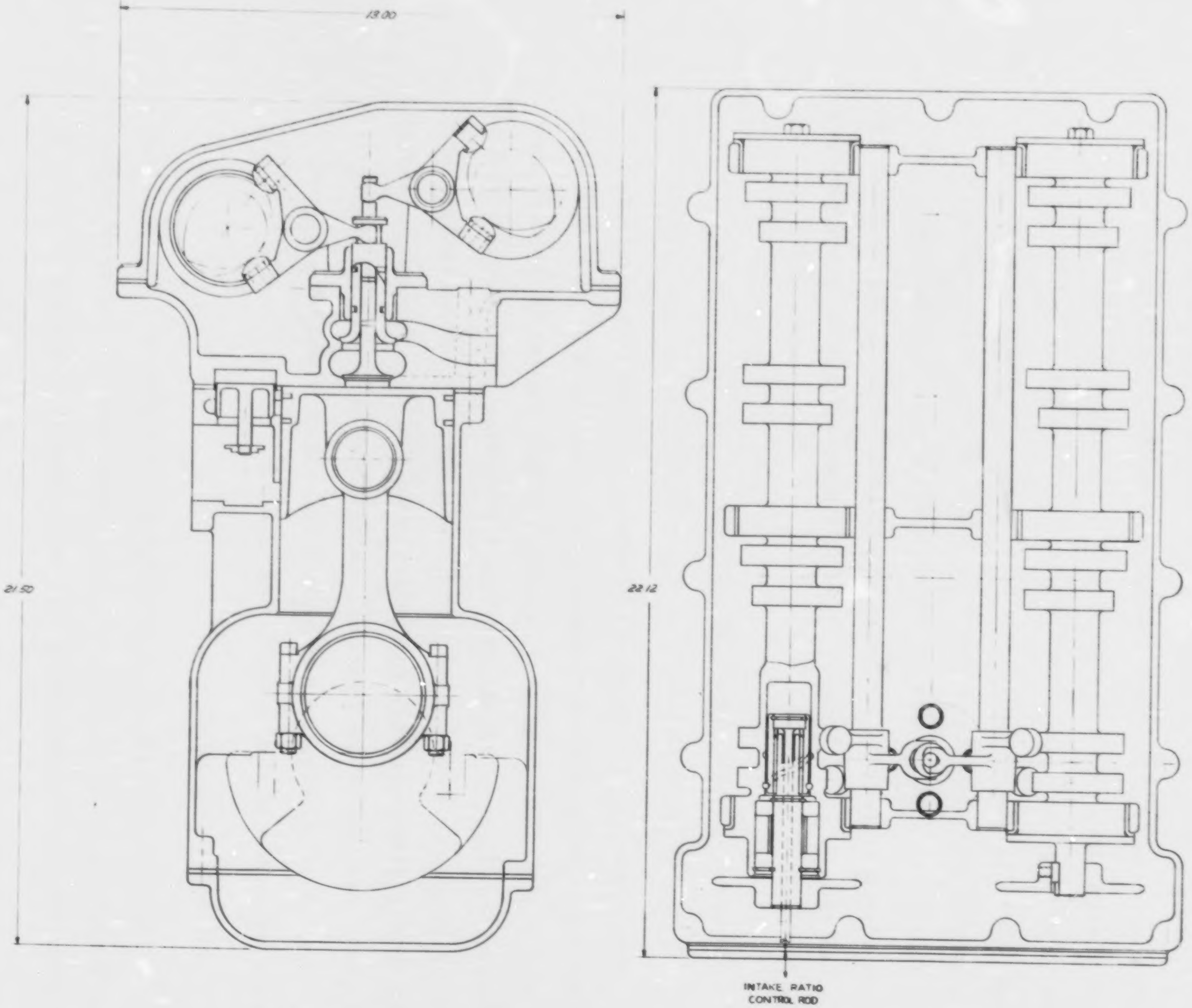

13.00
21.50
22.12
INTAKE RATIO
CONTROL ROD

Figure 4. 3.7 Mechanically Operated Intake Valving System.

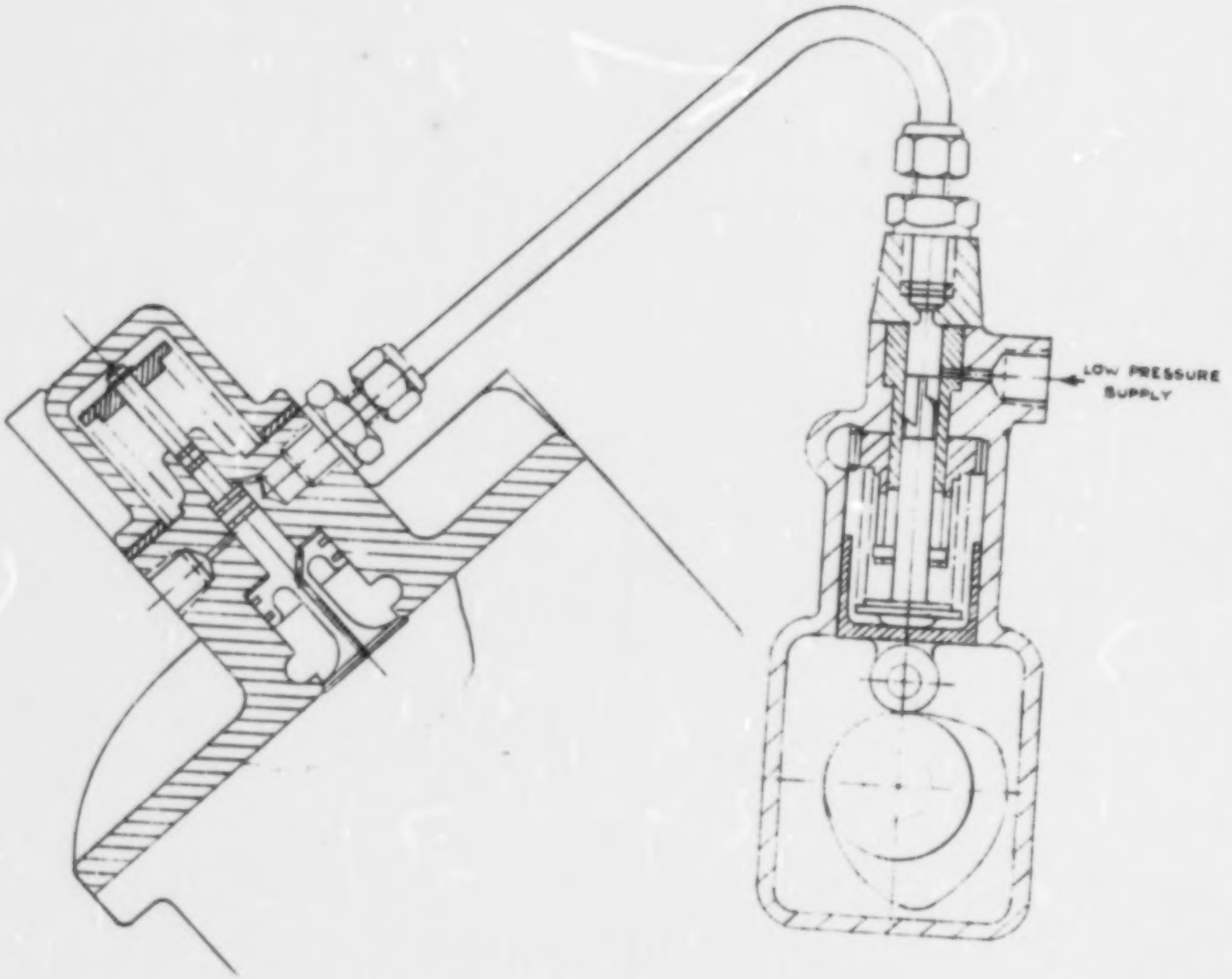

LOW PRESSURE SUPPLY

Figure 4. 3. 8 Directly Actuated Hydraulic Valve.

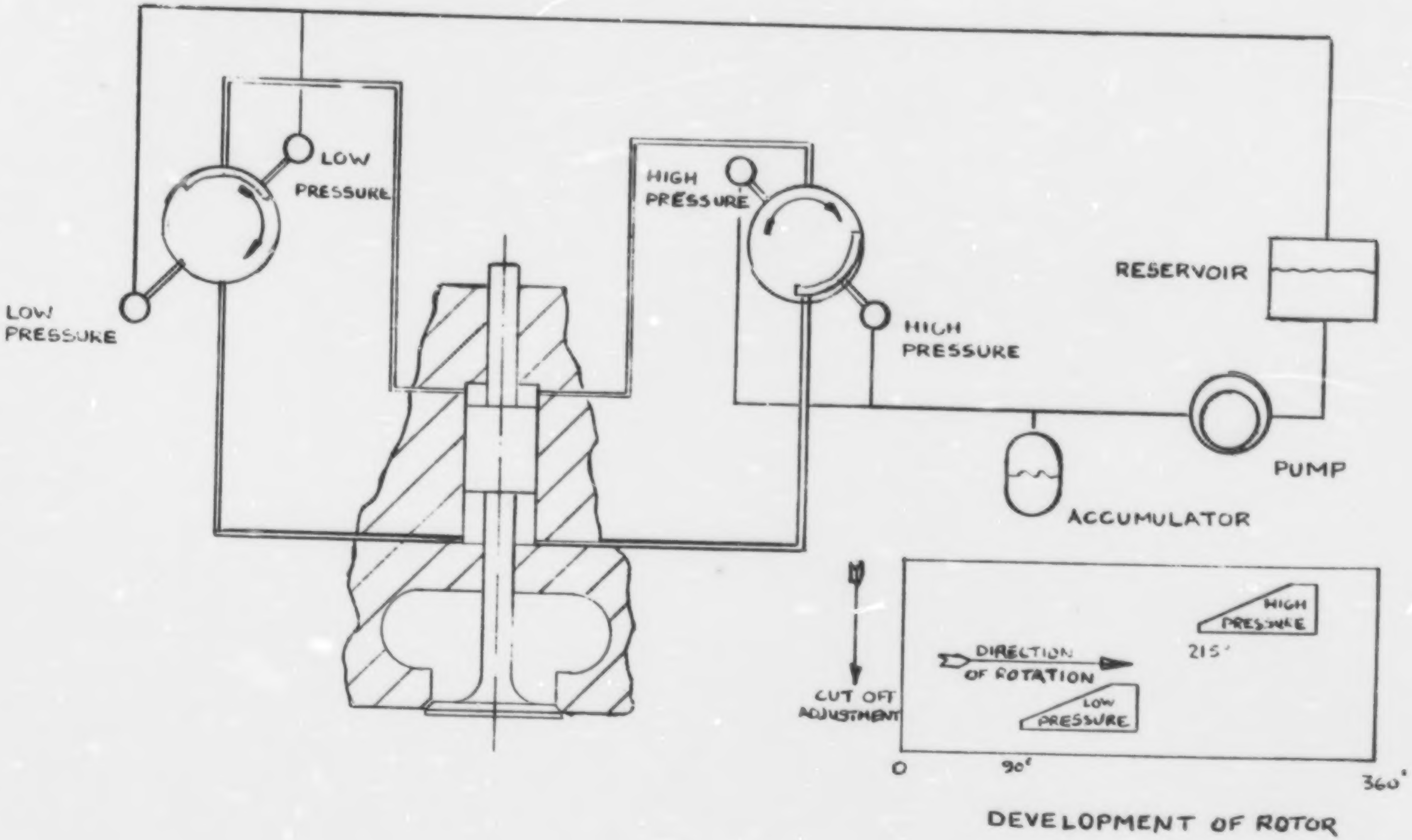

LOW PRESSURE
LOW PRESSURE
HIGH PRESSURE
HIGH PRESSURE
RESERVOIR
PUMP
ACCUMULATOR
CUT OFF ADJUSTMENT
DIRECTION OF ROTATION
HIGH PRESSURE
LOW PRESSURE
215°
0
90°
360°
DEVELOPMENT OF ROTOR

Figure 4.3.9 Pilot Operated Hydraulic Valve.

a piston which is solidly connected to the poppet valve. Thus, hydraulic pressure both opens and closes the valve, and variations in timing are achieved by sliding the rotary valve along its axis of rotation. There are a number of variants of this system, some of which are currently under development by injection equipment manufacturers as diesel engine fuel injection systems. This approach would also require a pressure balanced inlet valve, otherwise the pressure required to close the valve (or open it, if it opened outward) would become so high that the work required to operate the valves would become excessive. Preliminary calculations indicate that with an overall hydraulic efficiency of 50%, this system will require 2 to 3 hp at full output and 2000 rpm.

d. Recommended Approach

Detailed design studies of the most promising hydraulic system should be carried out, possibly with the aid of a manufacturer familiar with similar equipment. A mechanical system should also be examined in detail (probably the two inlet valves in series approach). The most promising of these two alternatives should then be constructed in the form of a bench test rig and developed to the fullest extent possible before being installed in an engine.

4.3.4 Expander Exhaust Valving

The exhaust valving is completely automatic, requiring no cams or other actuating means. The method of operation is shown schematically in Figure 4.3.10. The exhaust valve for the actual expander is shown in Section C-C of Figure 4.3.12.

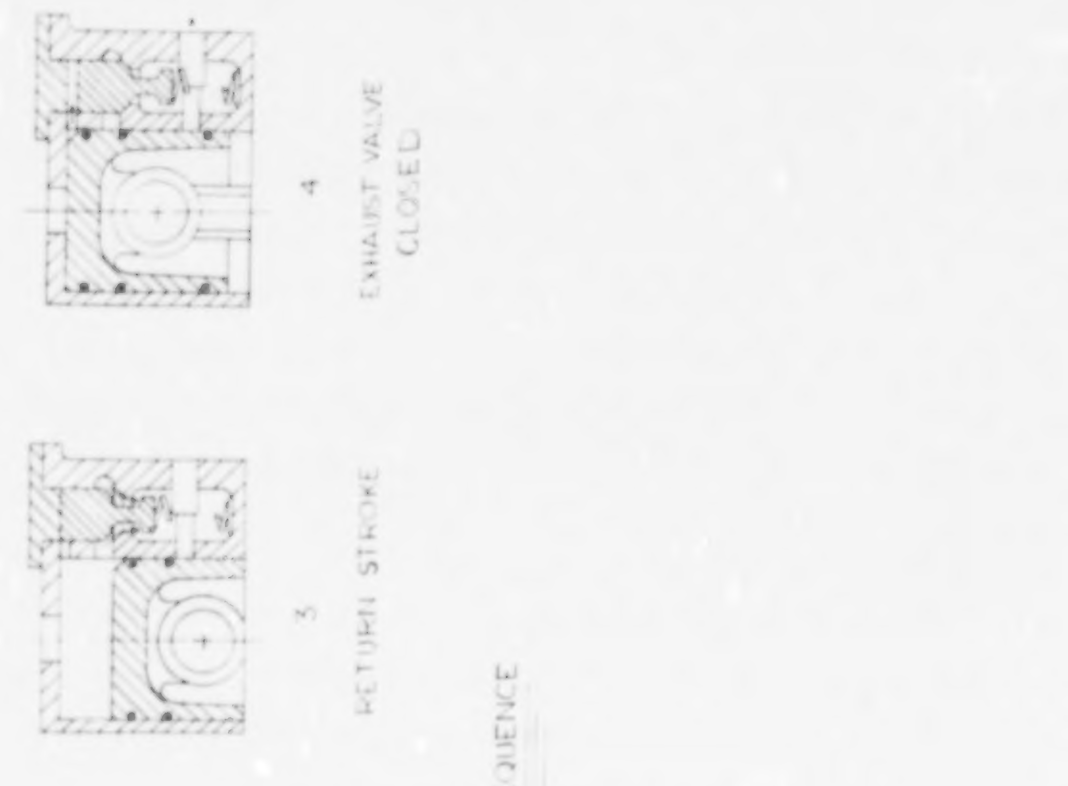

1
POWER STROKE
2
EXHAUST VALVE
OPENED
3
RETURN STROKE
4
EXHAUST VALVE
CLOSED
EXHAUST VALVE SEQUENCE
PRESSURE
DISPLACEMENT
1
2
3
4
INDICATOR DIAGRAM

Figure 4.3.10 Schematic of Exhaust Valve Function.

4.3.5 Engine Bearings

A detailed analysis of the engine journal bearing loading was undertaken. The connecting rod big end bearing was selected for analysis since its loading is more severe than the main bearings. The piston pin bearing is also severely loaded, but an analysis of this bearing was not undertaken, due to the lack of analytical technique for studying bearings where operation is almost entirely dependent on squeeze and partial film effects.

Preliminary analysis at both high and low speeds showed that the 300 rpm, maximum torque condition (80% intake ratio) would result in the most severe bearing loading condition. This should be contrasted with the internal combustion engine, where minimum oil film thicknesses in journal bearings usually occur at high speed during the intake or exhaust stroke and are due to inertia forces alone in almost all modern engines.

The oil film thickness was calculated as a function of crank angle and is plotted in Figure 4.3.11. Shown along with the Thermo Electron analysis is an analysis by Clevite Corporation, a major supplier of automotive engine bearings. Thermo Electron used a 0.002-inch diameter clearance in their analysis, Clevite a 0.003-inch diameter clearance; the Clevite figure is probably more realistic. Both analyses indicate minimum oil film thicknesses considerably less than the 100 microinches generally considered to be a safe minimum. An analysis of the 302 in^3 Ford V-8 connecting rod bearing at full throttle and 3000 rpm predicts a minimum oil film thickness of about 83 microinches, whereas the Clevite analysis predicts a minimum of 30 microinches for the Rankine-cycle engine.

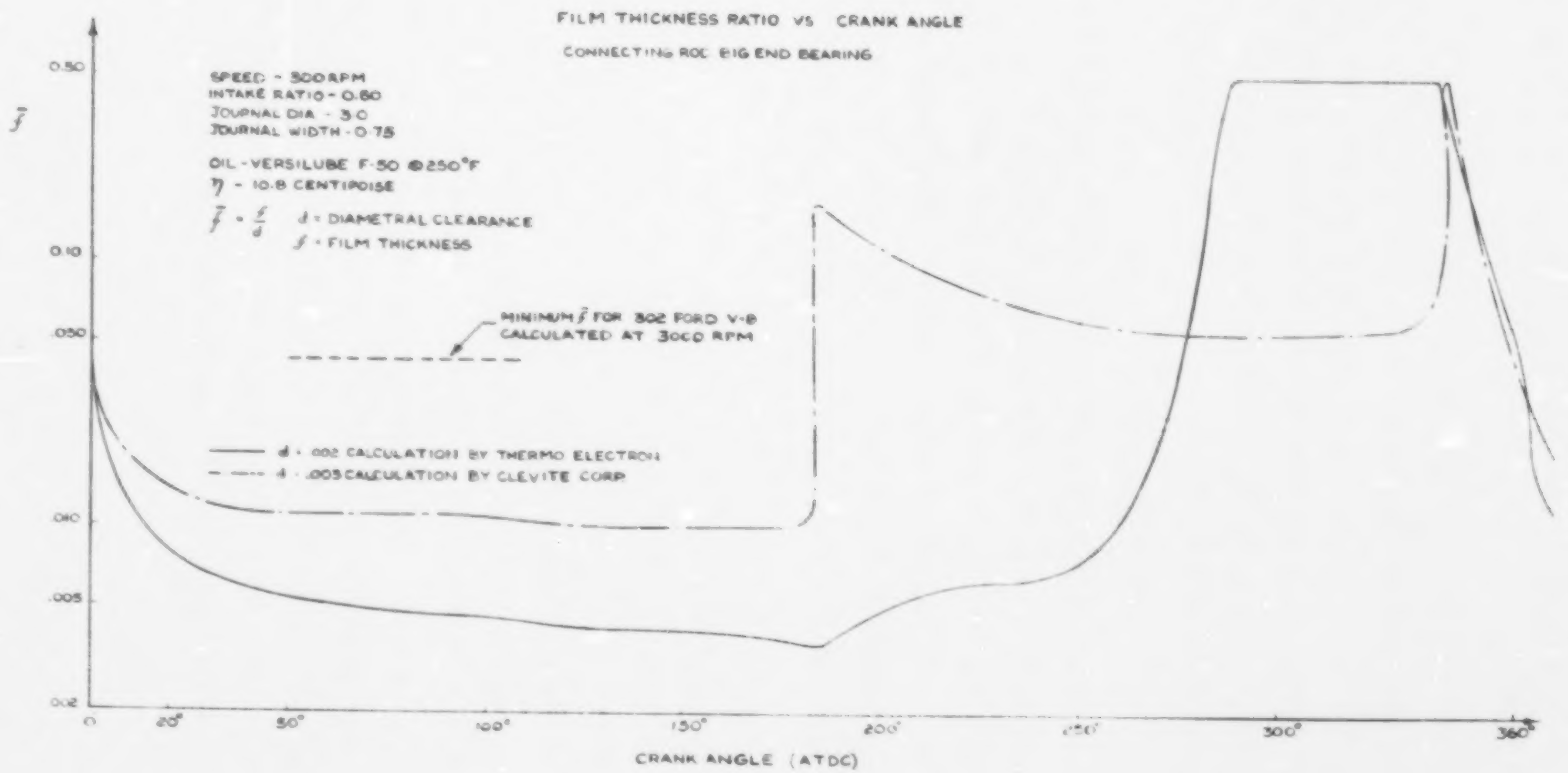


Figure 4.3.11 Oil Film Thickness as a Function of Crank Angle.

Reduction of the maximum intake ratio from 80% to 29% should improve the situation. If a fluid coupling is used rather than a clutch, there should be very little bearing problem, since the engine must speed up considerably before very much torque can be extracted. The use of a two-speed transmission should also alleviate the bearing loading considerably, particularly if the maximum intake ratio is limited to 0.29. The analysis shows that the bearings are quite conservative for higher speed operation.

It would probably not be practical to increase the diameter of the bearings since the connecting rod would not go out through the bore; this would cause severe assembly problems. The bearings could be increased in length, but the engine would undoubtedly become too long if the bearings were designed to give a minimum film thickness of 100 microinches at 300 rpm and 80% intake ratio.

Journal bearings are not feasible on an engine directly connected to the driveshaft, since the engine must start and carry heavy torque at practically zero rpm. In this case, anti-friction or roller bearings must be used.

The above analysis was carried out assuming that the lubricant is pure oil. Under start-up conditions, special precautions must be taken to ensure that this is the case, since the oil and thiophene are completely miscible. This is accomplished by heating the oil prior to start-up with the hot exhaust from the burner. In this way, any thiophene would be boiled out of the lubricating oil, thus ensuring a good supply of high viscosity lubricant to the bearings. A more complete discussion of this system is given in Section 4.11.

The first prototype expander should be constructed with caged needle bearings rather than with hydrodynamic bearings so that problems in the lubricating system may be solved without risking seizure of the expander. Analysis shows that this type of bearing should function quite well in the same geometry designed for the journal bearings.

4.3.6 Final Expander Design

The final expander design is shown in Figures 4.3.12 to 4.3.15. It is shown equipped with a hydraulically actuated valve of the type shown in Figure 4.3.8. The valve actuating pump and feedpump are driven off the front of the expander (see Figure 4.3.14) and are in direct communication with the expander crankcase. The single shaft seal, which is described more fully in Section 4.9.2, is just aft of the rear main bearing. The engine is shown coupled to the Dana single-speed transmission. The accessory drive is located in the rear bell housing, as shown in Figure 4.3.15. Casting thicknesses and crankcase design conform to current automotive practice (the peak cylinder pressure of 500 psi is roughly equivalent to that of IC engines). Three-cornered sealing surfaces, which are used in most IC engines, are avoided here because seal joints, particularly in the crankcase, are subjected to substantially higher pressure differentials than occur in IC engines and must be vacuum tight.

All of the major expander components are cast iron; the crankshaft would be surface-hardened to provide good bearing surfaces and would be counterweighted at both ends to eliminate the primary unbalanced moment. Static seals would be iron with molded rubber "o-ring" type inserts.

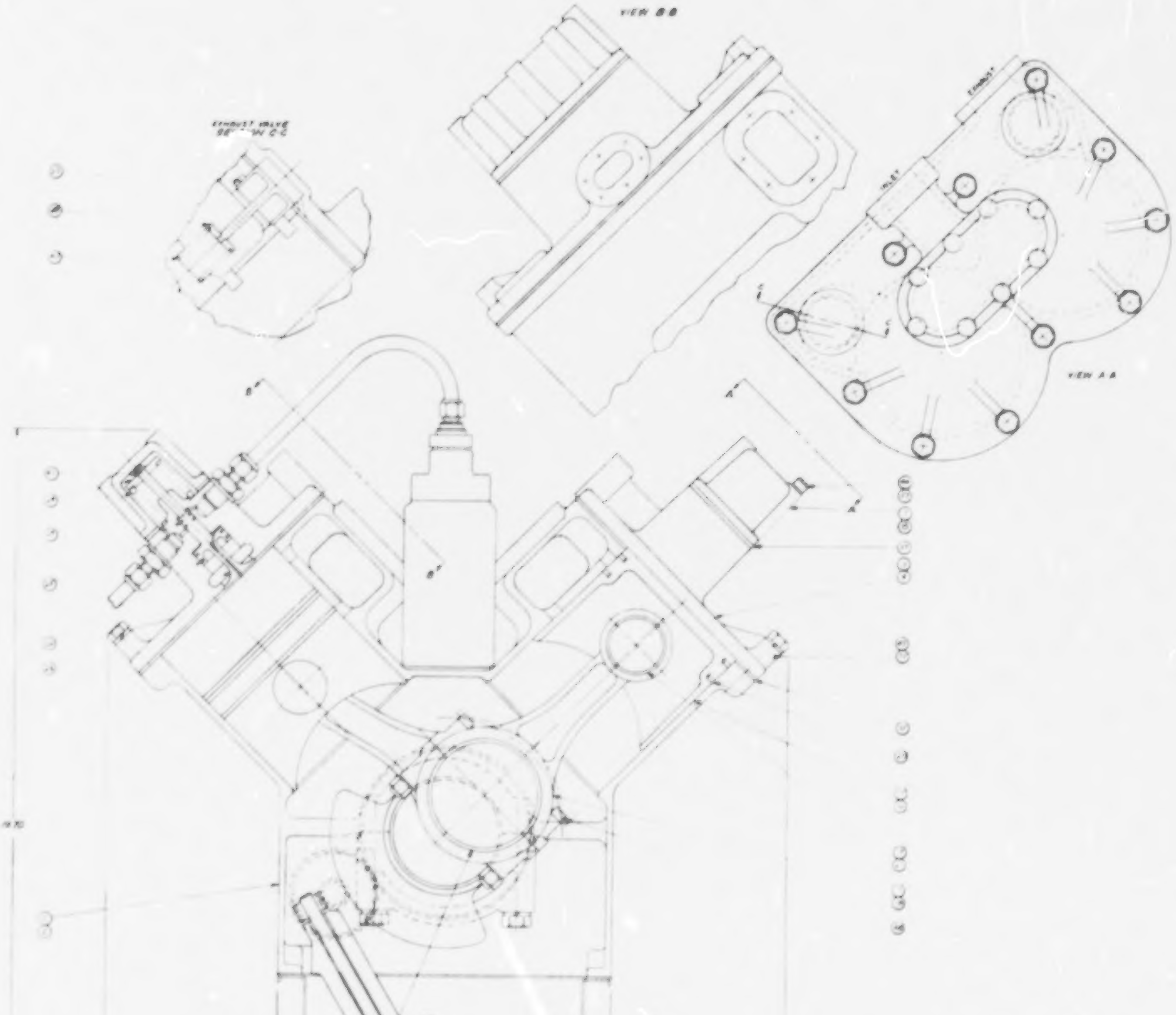

VIEW B B
EXHAUST VALVE
INLET
EXHAUST
VIEW A A

Figure 4. 3.12 V-4 Expander with Hydraulically Actuated Valves, Front View.

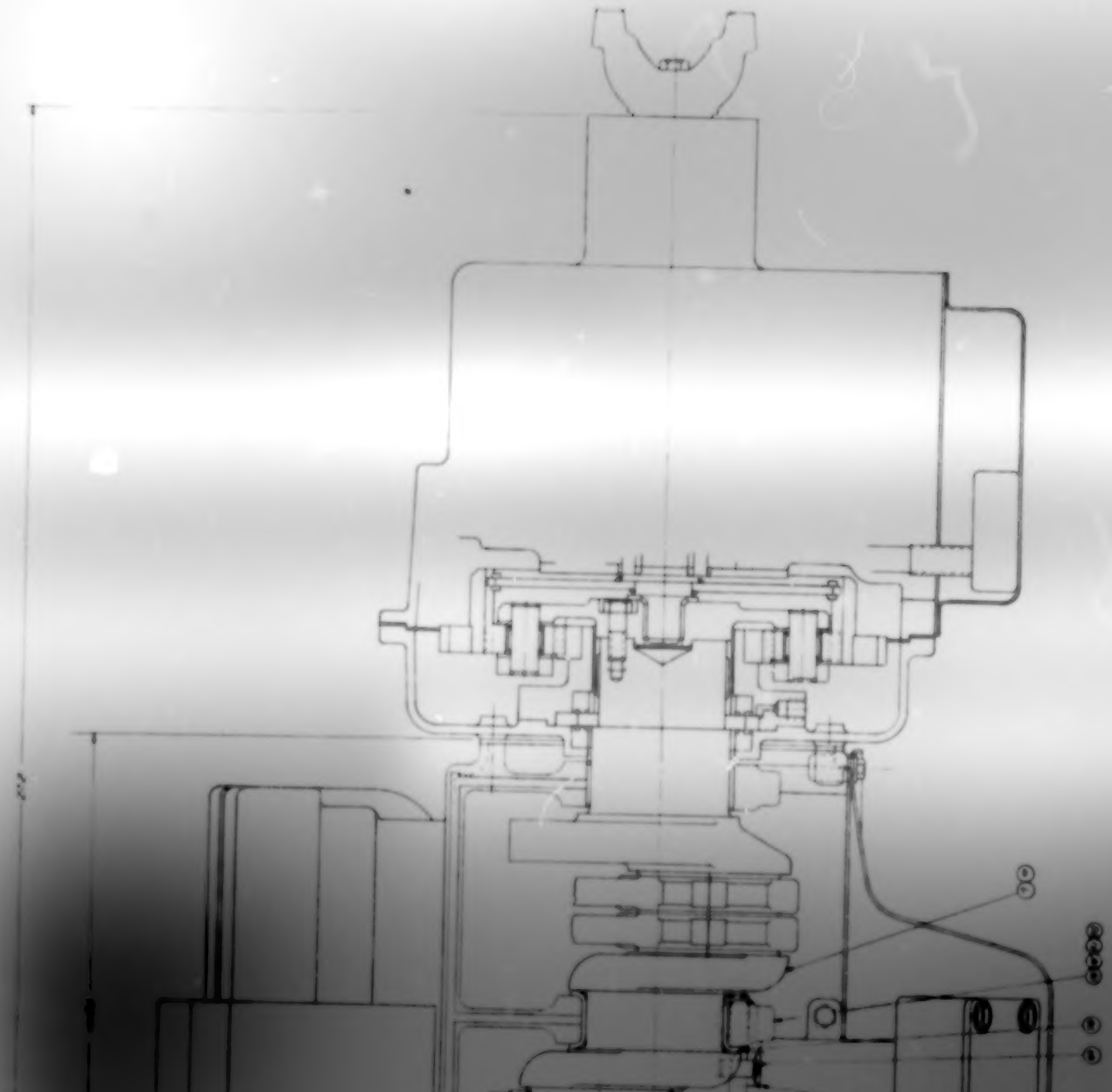


Figure 4.3.13 V-4 Expander with Hydraulically Actuated Valves, Side View.

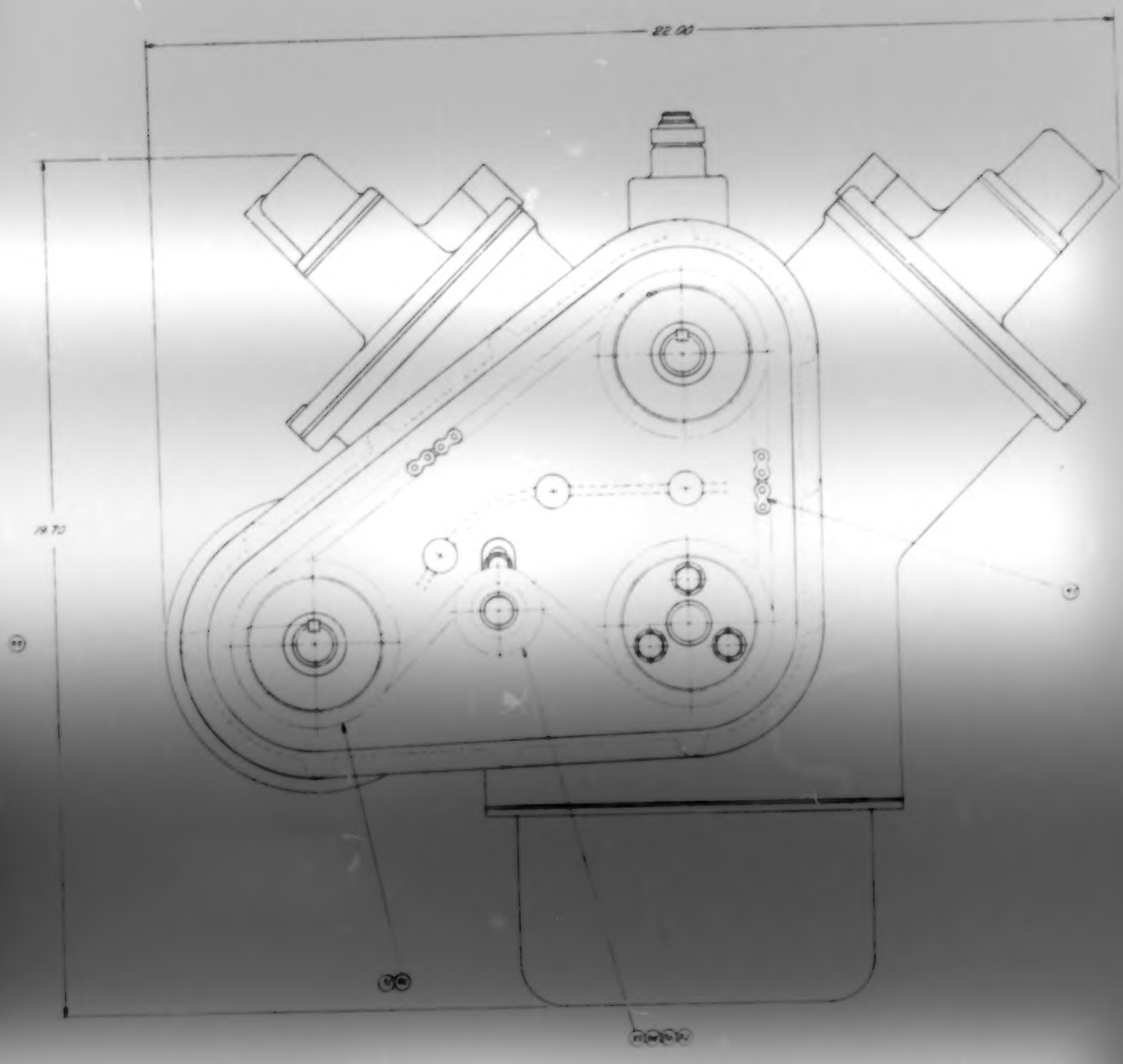


Figure 4.3.14 V-4 Expander with Hydraulically Actuated Valves, Showing Feedpump and Valving Drive.

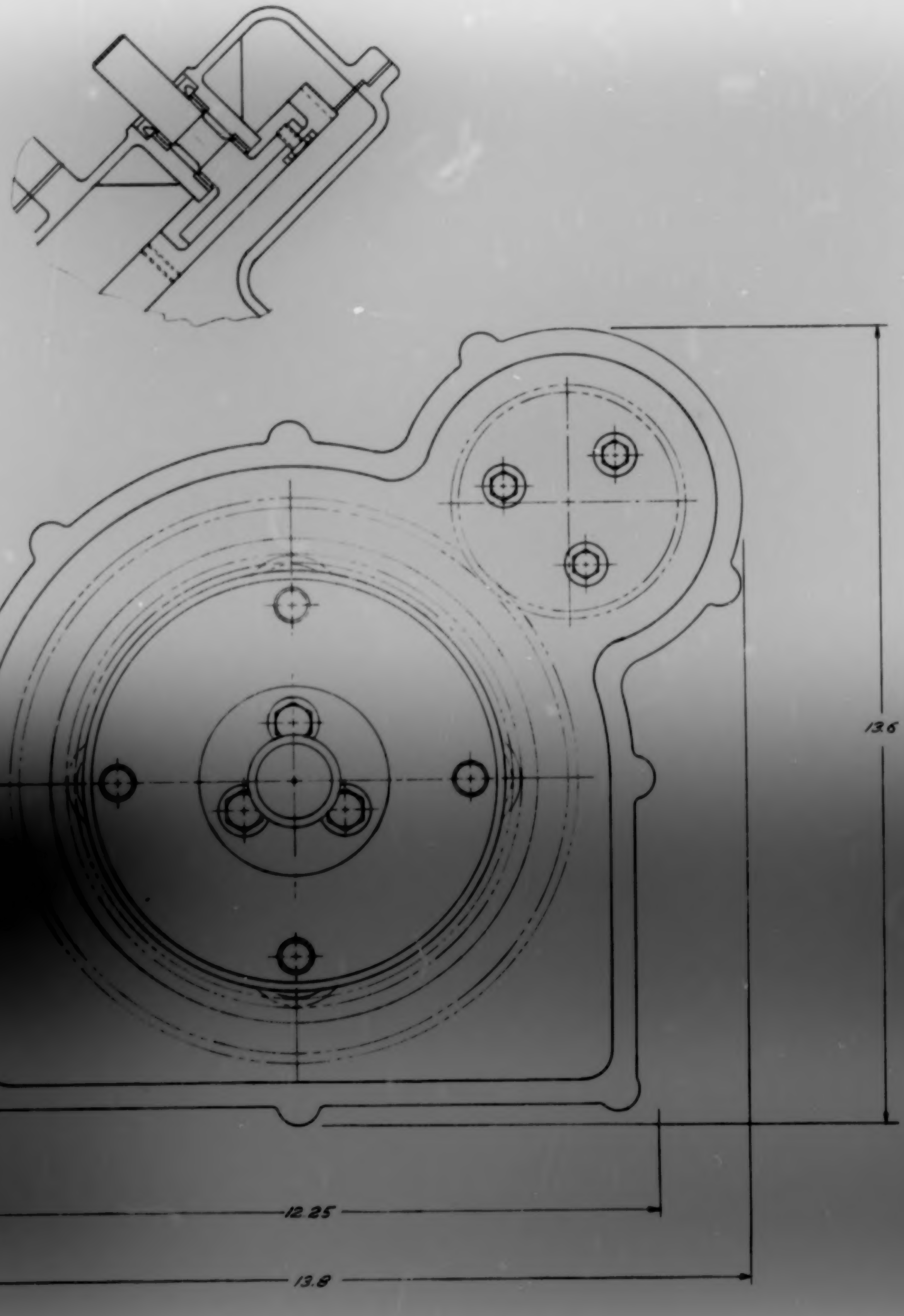


Figure 4.3.15 V-4 Expander with Hydraulically Actuated Valves, Showing Accessory Drive from Transmission End.

Major problem areas in the expander are as follows:

1. To develop a reliable valving system which will provide good flow area over the required range of intake ratios.
2. To ensure that the lubricant delivered to the bearings is of adequate viscosity, i.e., that it has a minimum of thiophene dissolved in it.
3. To provide a reliable shaft seal.
4. To establish some special quality control of castings in mass production to ensure vacuum tight expander assemblies.

4.4 FEEDPUMP DESIGN

Thermo Electron Corporation has tested several different types of positive displacement pumps with thiophene. Based on this testing, piston pumps are the only type presently giving satisfactory performance with thiophene for use in a Rankine-cycle system. The proposed feedpump is therefore a positive displacement piston pump with five cylinders driven by a wobble plate. The design drawings for pumps of three different displacements are illustrated in Figures 4.4.1 through 4.4.3, and their characteristics are summarized in Table 4.4.1. Since the feedpump must be able to supply 15 gpm pumping rate over a range of main engine speeds, the size of the feedpump required depends on how the feedpump is driven and the relationship between main engine rpm and feedpump rpm.

The organic flow rate to the boiler must be varied in response to the vapor demand in order to maintain constant boiler outlet pressure. Since the required pumping rate at any given engine speed can vary from zero to the maximum rate, depending on the intake ratio setting, it is necessary that the feedpump have variable displacement, permitting proper adjustment of the organic flow rate with no loss in efficiency. The method used to obtain variable displacement is similar to that used in diesel fuel injection pumps in which rotation of a piston with ramp undercut is used to vary the effective displacement of the pump. Rotation of all five pistons simultaneously is obtained by means of a central gear which meshes with gear teeth in the piston skirt; the central gear is controlled by a rack and pinion drive passing external to the pump through a rolling diaphragm hermetic seal

A computer analysis of the pressure transients produced by the flow ripple of piston pumps was carried out. This analysis indicated that at least 5 cylinders were required to limit the suction side transients so that cavitation on the pump suction did not occur. The net positive suction head to the feedpump is provided by subcooling of the liquid to the feedpump.

The pump could have been either crank-driven or wobble-plate driven. A wobble-plate drive was selected for the following reasons: compactness, easier integration with the engine, lower weight, less vibration, quieter operation, and more convenient geometry for incorporating variable displacement.

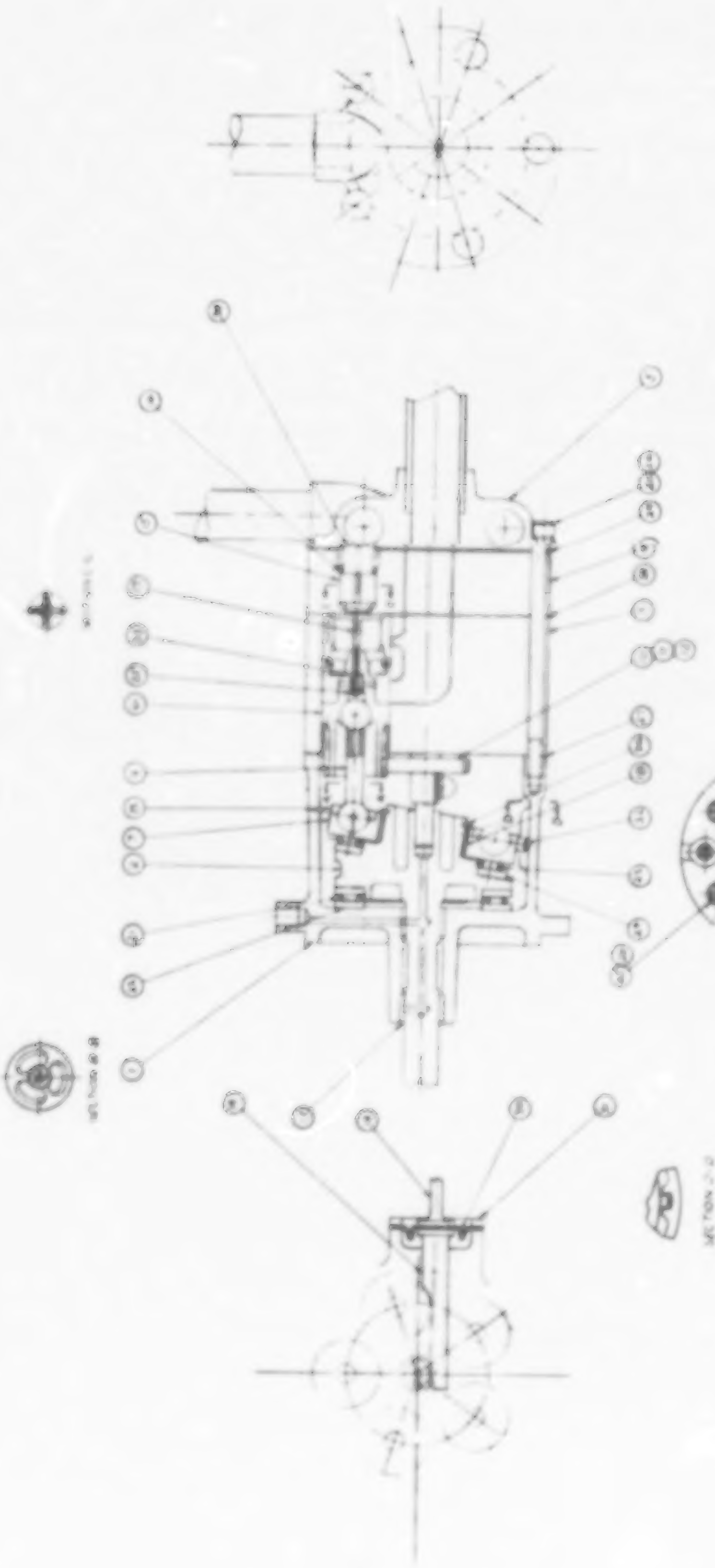


Figure 4.4.1 2000 rpm Feedpump.

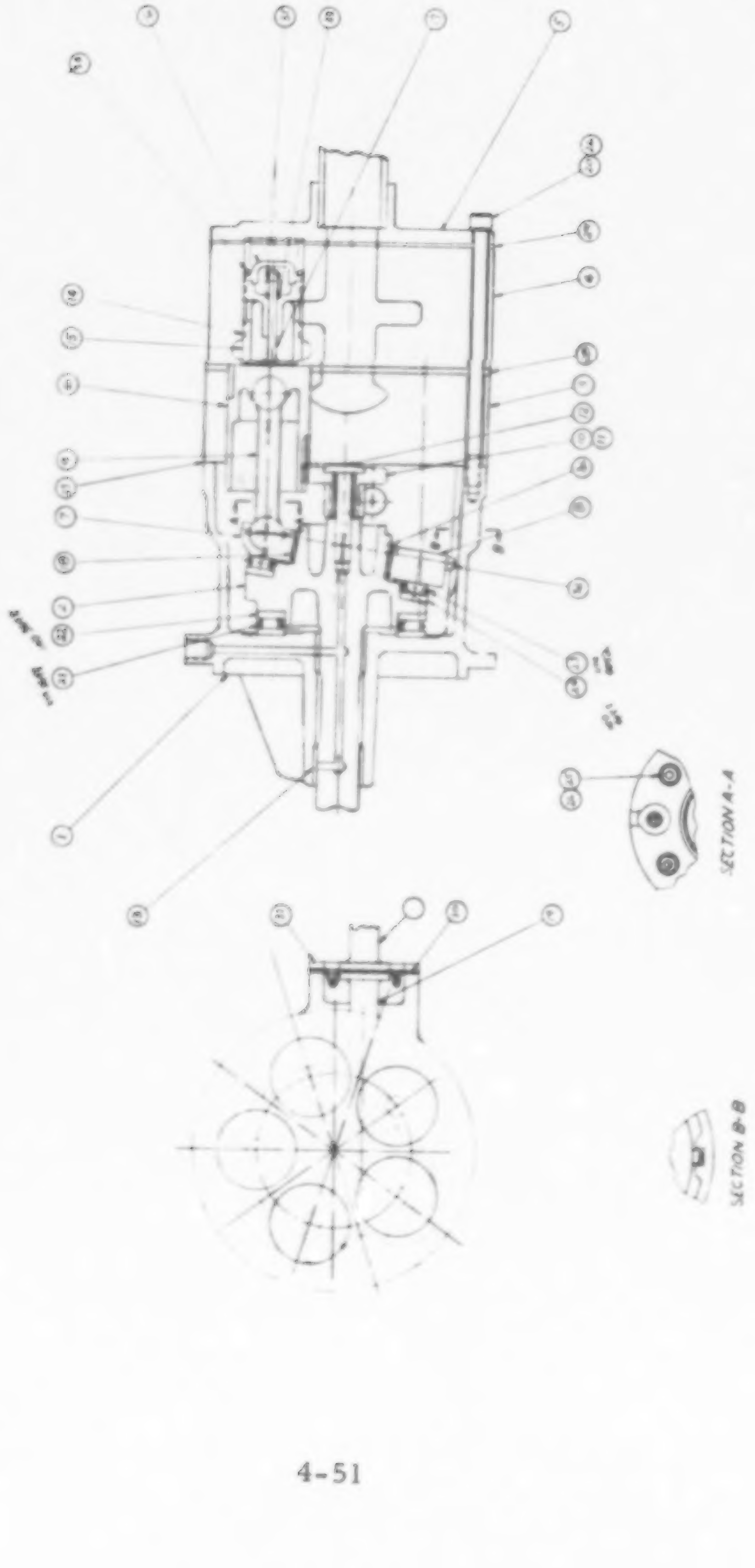

SECTION A-A
SECTION B-B

Figure 4.4.2 800-2000 rpm Feedpump.

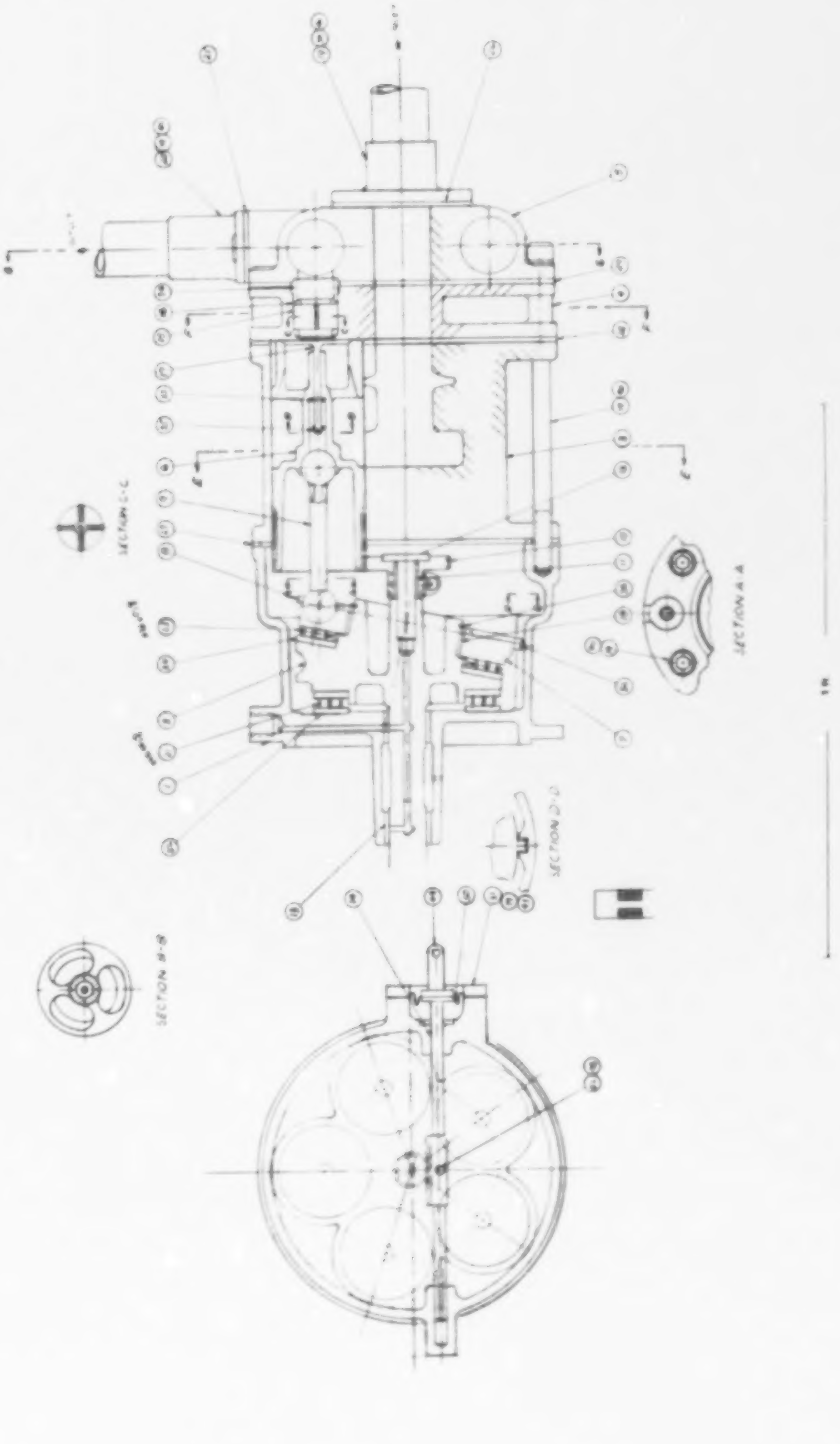

SECTION C-C
SECTION A-A
SECTION B-B
SECTION D-D

Figure 4.4.3 300-2000 rpm Feedpump.

TABLE 4.4.1

CHARACTERISTICS OF FEEDPUMPS FOR 15 gpm PUMPING RATE

Volumetric Efficiency = 90%, 5 cylinders

Pump Illustrated in Fig. No.	rpm Range For Maximum Pumping Rate	Total Displacement	Bore in	Stroke in	System for Which Pump Would be Used
4.4.1	2000 rpm	1.91	1.175	0.352	Driven at constant speed by auxiliary engine.
4.4.2	800-2000 rpm	4.78	1.595	0.478	Driven by main engine with maximum intake ratio of 0.29, 1/1 speed ratio.
4.4.3	300-2000 rpm	12.78	2.08	0.750	Driven by main engine with maximum intake ratio of 0.80, 1/1 speed ratio.

4.5 COMBUSTOR DESIGN AND CHARACTERISTICS

The combustor requirements are:

a. Low emission level for NO, CO, and unburned hydrocarbons for all operating conditions.

b. Turndown ratio of 15 to 1

c. Low pressure drop in compact burner

d. High reliability, low maintenance, and low cost.

In this section, a description of the combustor design and its operating characteristics is presented. Emission levels measured from a 140,000 Btu/hr burner at TECO which indicate the low-emission potential of the Rankine-cycle system are also presented, as well as a brief description of the fuels available for use in Rankine cycle propulsion systems.

4.5.1 Combustor Design and Fuel/Air Supply

The combustor design of this study is based on parametric information provided by the Marquardt Corporation, Van Nuys, California, under subcontract to Thermo Electron. The information is derived from measurements and calculations on the Marquardt SUE, or sudden expansion burner, illustrated schematically in Figure 4.5.1. In Figure 4.5.2, the predicted SUE burner characteristics are presented. This parametric plot indicates quantitatively the effect of combustion chamber diameter on pressure drop and combustion chamber length for a burning rate of 2.0×10^6 Btu/hr and equivalence ratios of 0.6, 0.8 and 1.0. The combustion chamber volume in this plot is based on a burning density of 2.8×10^6 Btu/hr-ft^3 at 1 atmosphere pressure. Marquardt has obtained limited experimental data which agree with the plot and the experimental

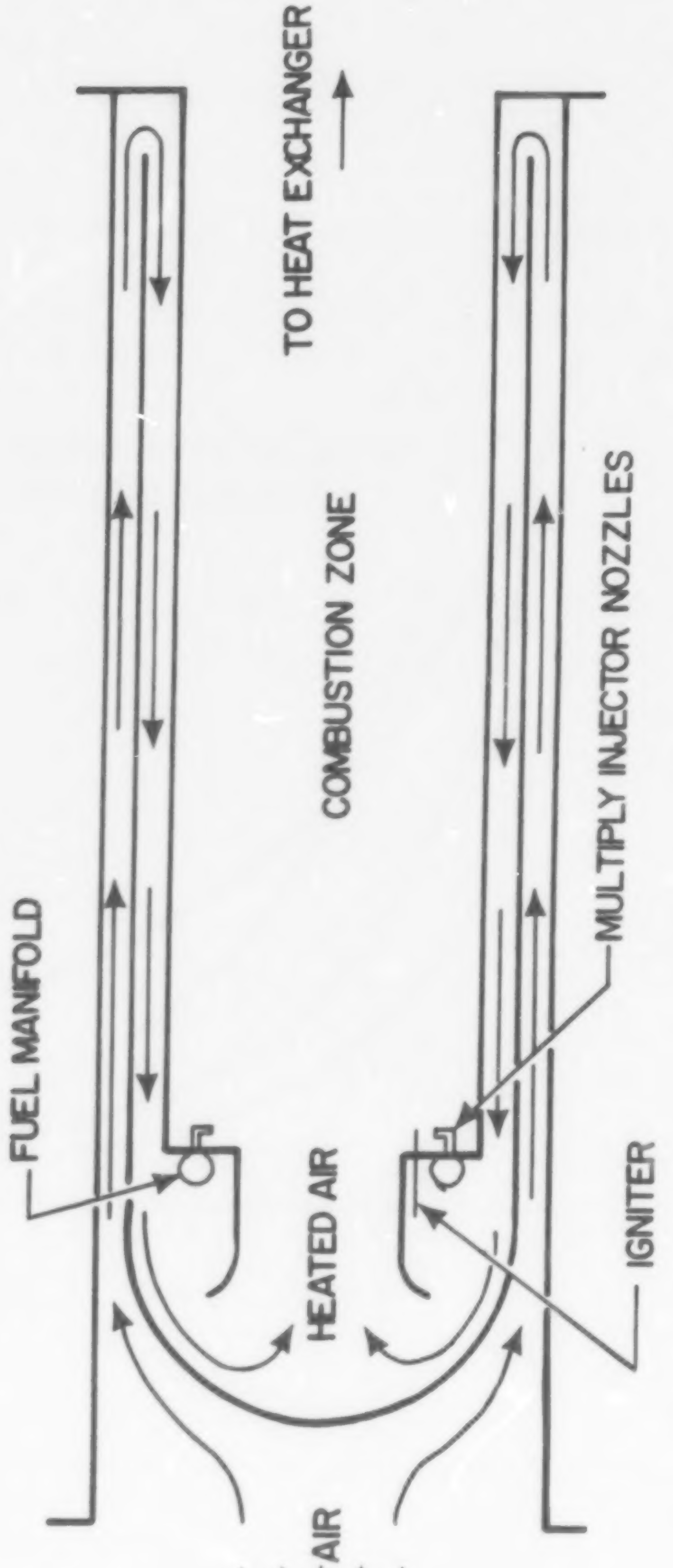

TO HEAT EXCHANGER
COMBUSTION ZONE
FUEL MANIFOLD
MULTIPLY INJECTOR NOZZLES
HEATED AIR
IGNITER
AIR

Figure 4.5.1. Marquardt Sudden Expansion Burner (SUE)

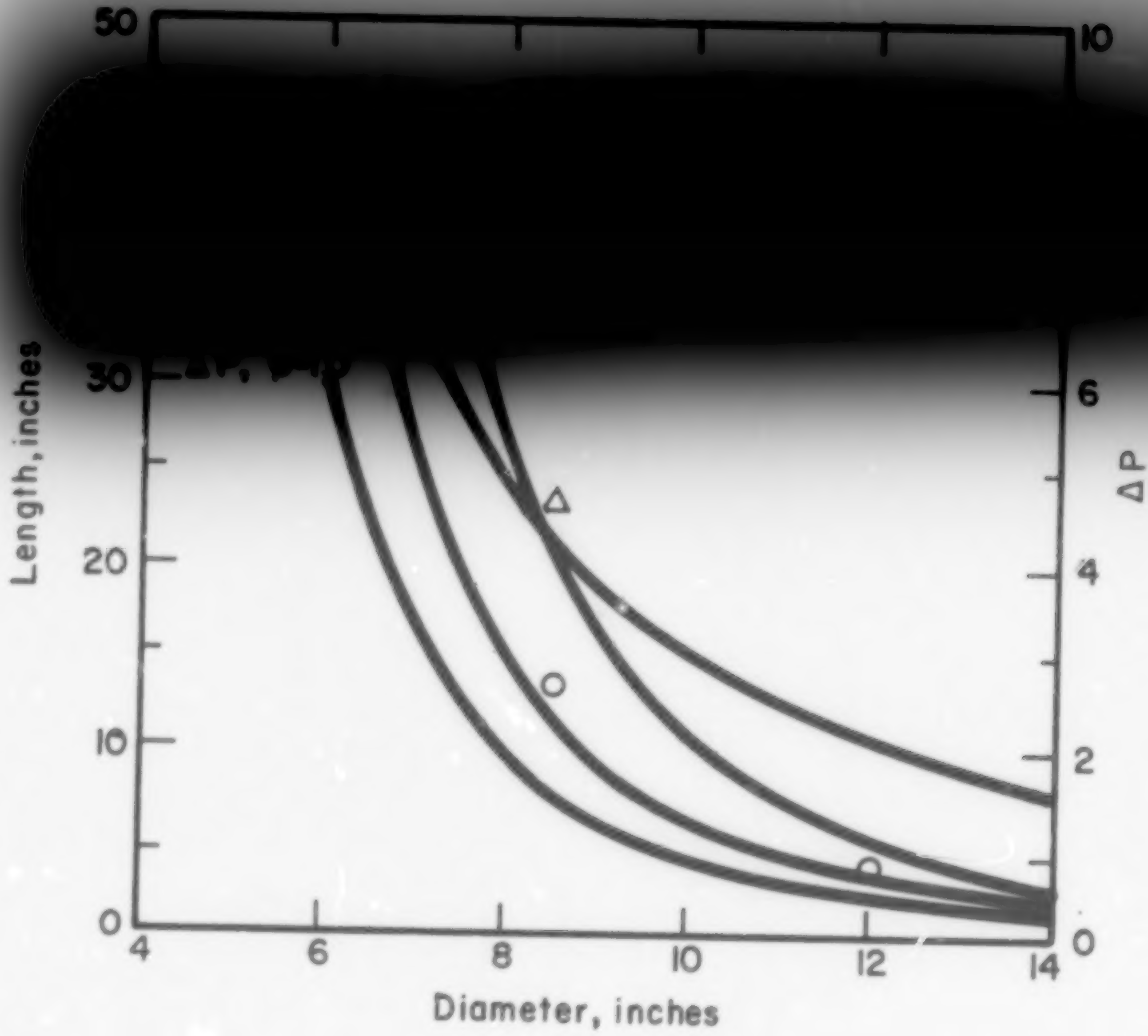


Figure 4.5.2 Predicted SUE Burner Characteristics.

points are indicated. Marquardt has also obtained burning densities up to 2.5 x 10^6 Btu/hr-ft^3 in a 500,000 Btu/hr version of this burner. Thus, this parametric plot should provide a reasonably accurate method for sizing of the burner. Scaling to different burning rates can be made, since, for a given burner diameter, the ΔP varies as the square of the burning rate and the burner length varies directly with the burning rate.

[illegible] shape for integration with the boiler, (2) use of a single fuel nozzle using compressed air for atomization, and (3) use of swirl vanes on the inlet nozzle to provide better fuel-air mixing and more stable combustion.

In Figure 4.5.3, cross-sectional views of the automotive-size burner are presented, and in Table 4.5.1 the design point burner characteristics are presented. The burner shape is designed to conform to the boiler tube bundle shape and sits directly on top of the boiler tube bundle. Two combustion chambers, each with a maximum burning rate of 1.05 x 10^6 Btu/hr, are used in parallel to provide a pressure drop of only 1.5 inches w.c. at full burning rate within the allowable combustion chamber diameter of 7.0 inches. The maximum burner diameter is restricted because of packaging considerations. It is essential to maintain a minimum combustion side pressure drop through the burner-boiler combination, since the combustion system must be operated electrically on startup and it is desirable to run the combustion system full-out on startup to minimize startup time. Care will be required, however, to insure that instabilities in the operation of the two parallel burners do not exist.

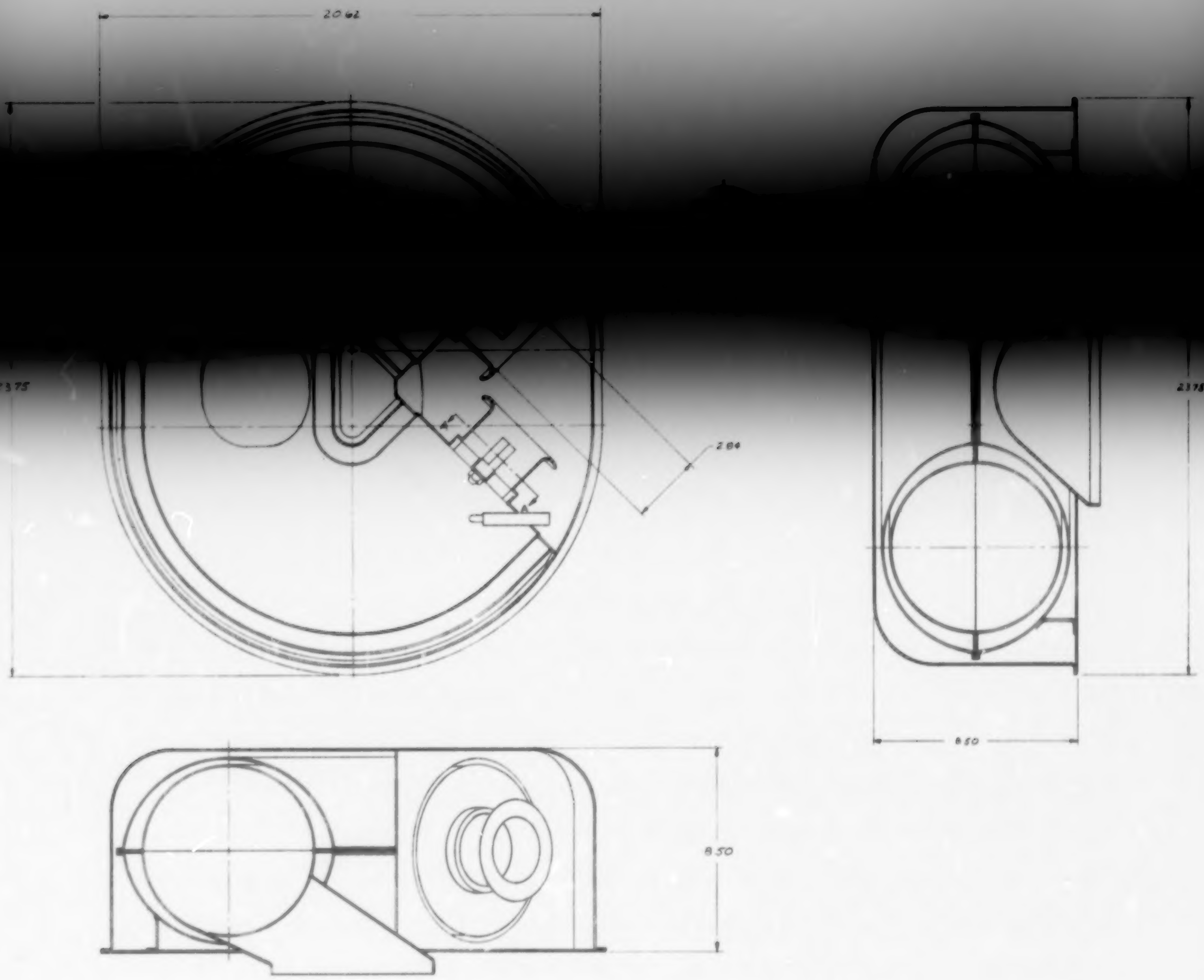

20.62
23.75
2.84
23.75
8.50
8.50

Figure 4.5.3 Cross Sections Through Automotive-Size Burner.

TABLE 4.5.1

BURNER CHARACTERISTICS

Maximum burning rate for each of two burners	1.05×10^6 Btu/hr
Combustion chamber diameter	7.0 in.
Maximum volumetric energy release rate	2.8×10^6 Btu/hr-ft^3
Combustion chamber length required	17.0 in.
Maximum pressure drop through burner	1.5 in. w.c.
Combustion chamber mass (22 Ga SS)	6.3 lbs
Air distributor mass (22 Ga CS, 20% free area)	5.4 lbs
Plenum cover mass (22 Ga CS)	8.2 lbs
Fastening hardware, nozzles, electrodes, strengthening struts, etc.	2.1 lbs
Total combustor mass	22.0 lbs
Heat transfer coefficient to burner wall (Measured by Marquardt Corporation)	$\frac{2.2 \text{ Btu}}{\text{hr-ft}^2 \text{ °F}}$
Combustor wall operating temperature	1500-1800°F
Combustion air temperature rise at inlet of nozzle	35-45°F
Average residence time in burner at maximum burning rate	12.5 millisec

During operation, combustion air from the blower is directed into the outer container which serves as a plenum. The air then flows through a perforated plate surrounding the combustion chamber and around the combustion chamber to the inlet end of the two burners. The combustion air is thus used to cool the combustion chamber wall; the distributor plate permits regulation of the air flow to eliminate potential hot spots.

The air flows through the nozzle into the combustion chamber through swirl vanes (Section A-A of Figure 4.5.3). Fuel is sprayed into the chamber by an air-atomizing nozzle; this type of fuel nozzle was selected for the following reasons:

a. It gives a finer fuel spray than pressure nozzles.

b. It can operate well over fuel flow rates from zero to rated capacity, i.e., it has a large turndown ratio.

c. It has a much larger fuel orifice than a pressure nozzle, insuring maximum freedom from clogging.

Combustion occurs and is completed in the combustion chamber, and the hot combustion products are then directed downward into the central plenum of the boiler tube bundle.

The requirements and characteristics of the fuel/air supply system are summarized in Tables 4.5.2 and 4.5.3. The maximum combustion-side pressure drop through both burner and boiler is 4.4 in w.c. The maximum blower horsepower is 0.60 hp and the compressor power 1/3 hp. For startup, the battery must supply 99 amps at 12 V dc to the combustion system. For normal operation, the alternator should supply 75.0 amps on the average to the combustion system.

TABLE 4.5.2

FUEL-AIR REQUIREMENTS AND CHARACTERISTICS

Fuel Composition (Assumed)	
Carbon	85%
Hydrogen	15%
Higher Heating Value (Calculated)	21,600 Btu/lb
Lower Heating Value (Calculated)	20,180 Btu/lb
Excess Air	33%
Air/Fuel Mass Ratio	19.8
Equivalence Ratio	0.752
Reference Cycle Burning Rate (82.7% HHV Boiler Efficiency)	1.91×10^6 Btu/hr
Maximum Burning Rate	2.10×10^6 Btu/hr
Fuel Flow Rate (Max. Burning Rate)	
lb/hr	97.2
gal/hr (ρ = 47.6 lb/ft^3 at 95°F)	15.3
Air Flow Rate (Max. Burning Rate)	
lb/hr	1921
CFM (95°F, 1 atm)	447

TABLE 4.5.3

FUEL/AIR SUPPLY SYSTEM

Combustion Air Blower

Required Pressure - Burner		1.50 in w.c.
- Boiler		2.90 in w.c.
Total		4.40 in w.c.
Blower Wheel (Torrington AA-610-314-2)		
Diameter		6.28 in.
Depth		3.46 in.
Weight (Aluminum)		0.48 lb
Speed		3800 rpm
Shaft Horsepower		
at 440 cfm		0.60 hp
at 0 cfm		0.28 hp
Blower Efficiency		
at 440 cfm		50%
Housing (see Figure (a) page 4-73)	A	10.61 in.
	B	9.76 in.
	M	4.75 in.
	K	3.79 in.
	Weight	~ 2.0 lbs

Motor Power Requirement

	440 CFM	Low CFM
Shaft Power	446 watts	208 watts
Electrical Power (60% Motor Efficiency)	745 watts	347 watts
Amperage at 12 V dc	62.0 amps	28.9 amps

Total Weight Including Motor 20.0 lbs

Fuel Atomizing Air

Pressure Required	9.5 psig
Air Flow - SCFM	4.6
Compressor Gast Oil-Less Model No.	0740
Speed	1725 rpm
Motor Size Specified	1/3 hp
Ideal Power Required	0.25 hp
Motor Shaft Power Requirement	248 watts
Motor Efficiency	60%
Electrical Input	414 watts
Amperage at 12 V dc	34.5 amps
Weight including Motor	22 lbs

Fuel Supply (Driven by Compressor Motor at 1725 rpm)

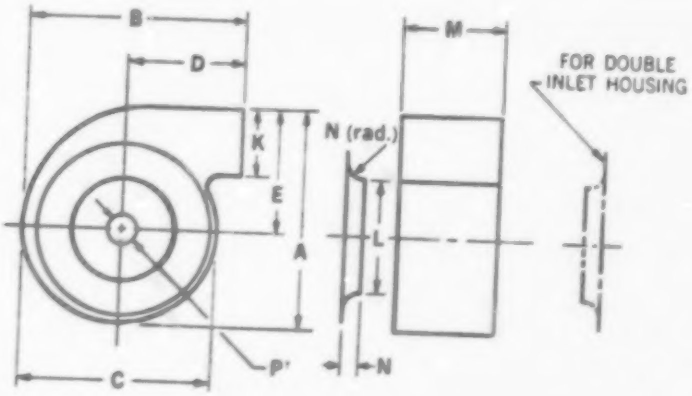
Pressure Required	25 psig
Pumping Rate	15 gph
Ideal hp	0.00364 hp
Shaft Power (50% Pump Efficiency)	0.00728 hp
Electrical Power (60% Motor Efficiency)	9.1 watts
Amps at 12 V dc	0.76 amps
Weight of Pump	0.65 lbs

Ignition and Flame Sensing

Ignition	Electrode Spark
Electric Power Requirement	12 watts
Amperage at 12 V dc	1.0 amps
Flame Sensing	CdS Cell
Electric Power Requirement	8.4 watts
Amperage at 12 V dc	0.7 amps
Total Weight	0.35 lbs

Total Electric Requirement for Combustion System	
Voltage	12 V dc
Total Amps	
Startup	99.0 amps
Operating-Peak	97.3 amps
Operating-Min	64.2 amps
Operating-Estimated Average	75.0 amps
Total Combustion System Weight	43 lbs

(a)


B
D
K
E
A
C
P'
N (rad.)
M
L
N
FOR DOUBLE INLET HOUSING

†P=FOR SINGLE INLET HOUSINGS

4.5.2 Emission Levels from Rankine-Cycle Burners

Of great importance is the level of emissions that potentially can be expected from Rankine-cycle automotive propulsion systems. While a full-scale burner for this system has not yet been constructed and tested, the experimental evidence available indicates strikingly low emission levels for NO, CO, and unburned hydrocarbons in an external combustion system without air preheat. In Figures 4.5.4 through 4.5.9 and Table 4.5.4, measured emission levels from a burner developed at TECO for use in a 3 kwe engine-generator set are presented, indicating extremely low emission levels under both steady-state and transient operation. This burner is 1/9th the size of each burner required for the automotive propulsion system. Also presented are burner-on and burner-off transients indicating that startup or shutdown of the system should not be troublesome.

The Marquardt Corporation has made some measurements on a 500,000 Btu/hr SUE burner, operating with primary air only, at an equivalence ration of 0.75. Typical steady-state results from this test are:(8)

Unburned hydrocarbons	4 ppm
CO	60 ppm
NO	90 ppm

Results presented by General Motors on the burner used in the GM SE-101 steam-powered car are illustrated in Figure 4.5.10. At the lowest fuel/air ratio used, at 60% of design air flow, UHC concentration is 8 ppm, NO is 75 ppm, and CO is 300 ppm. The GM burner has a quoted heat release rate of 3.4×10^6 Btu/hr, a pressure drop of 13.6 in w.c., and a burning density of about 5×10^6 Btu/hr-ft^3. This burner

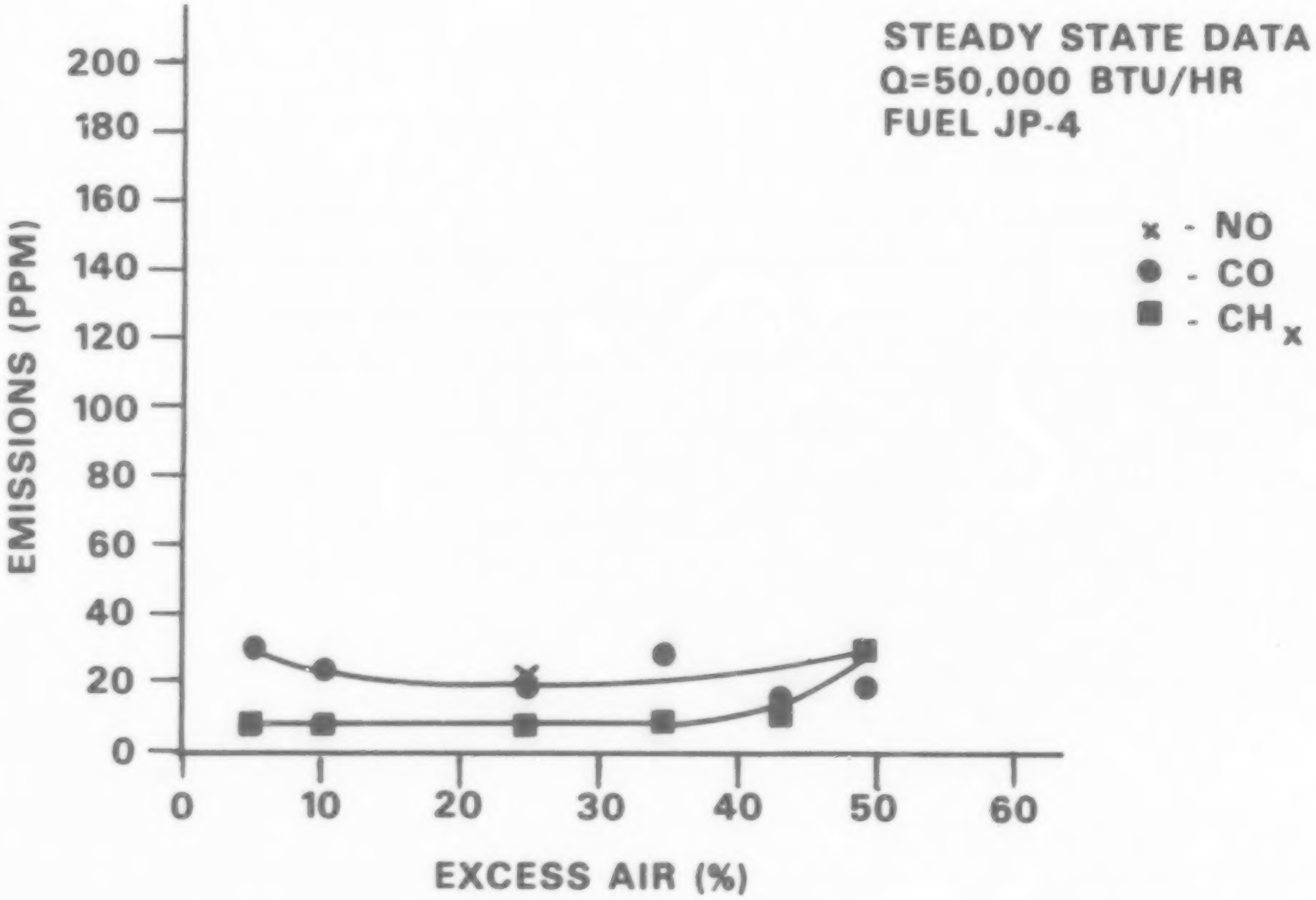


Figure 4. 5. 4 Effect of Excess Air on Emissions, 50,000 Btu/hr

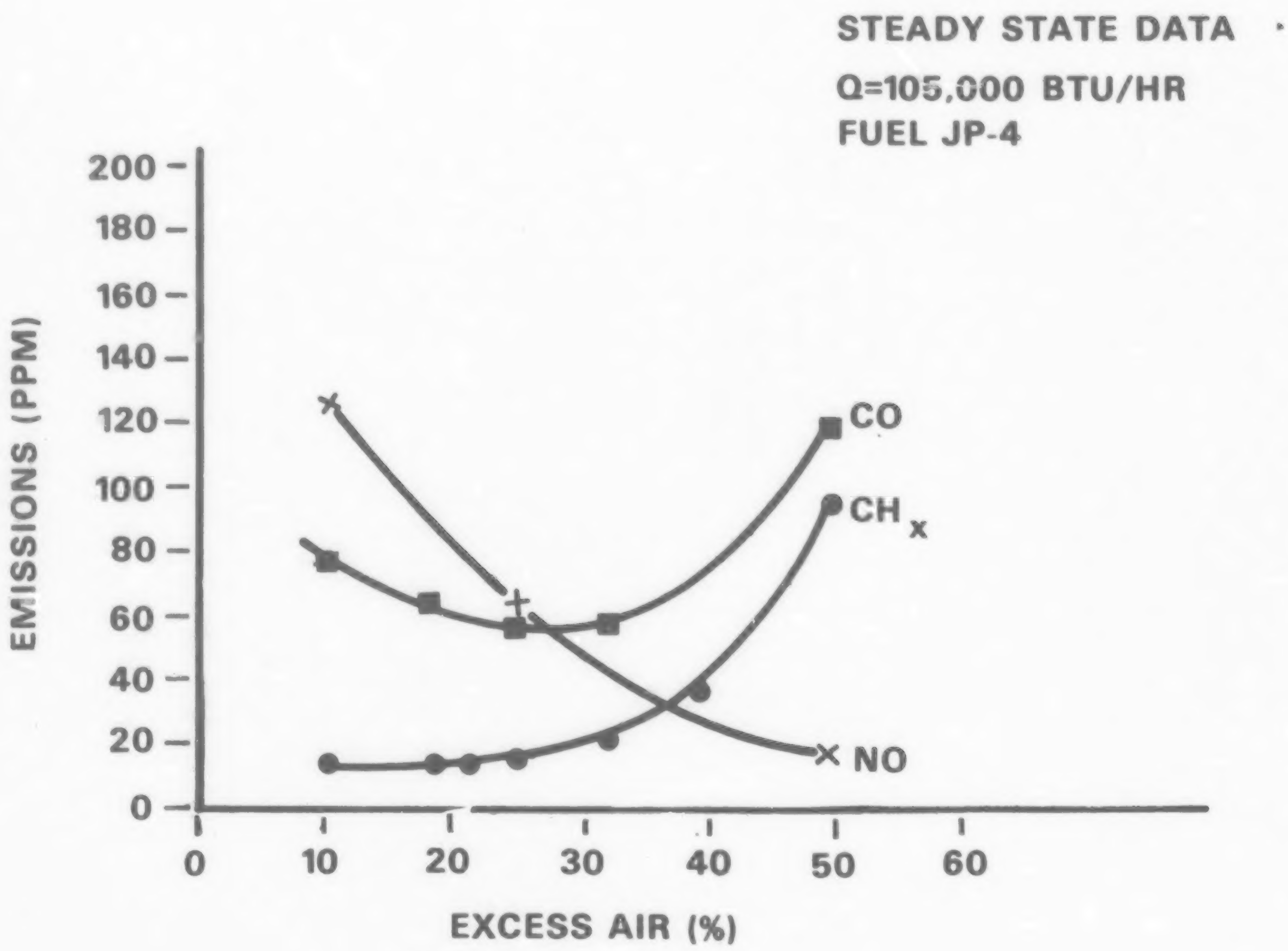


Figure 4. 5. 5 Effect of Excess Air on Emissions, 105, 000 Btu/hr

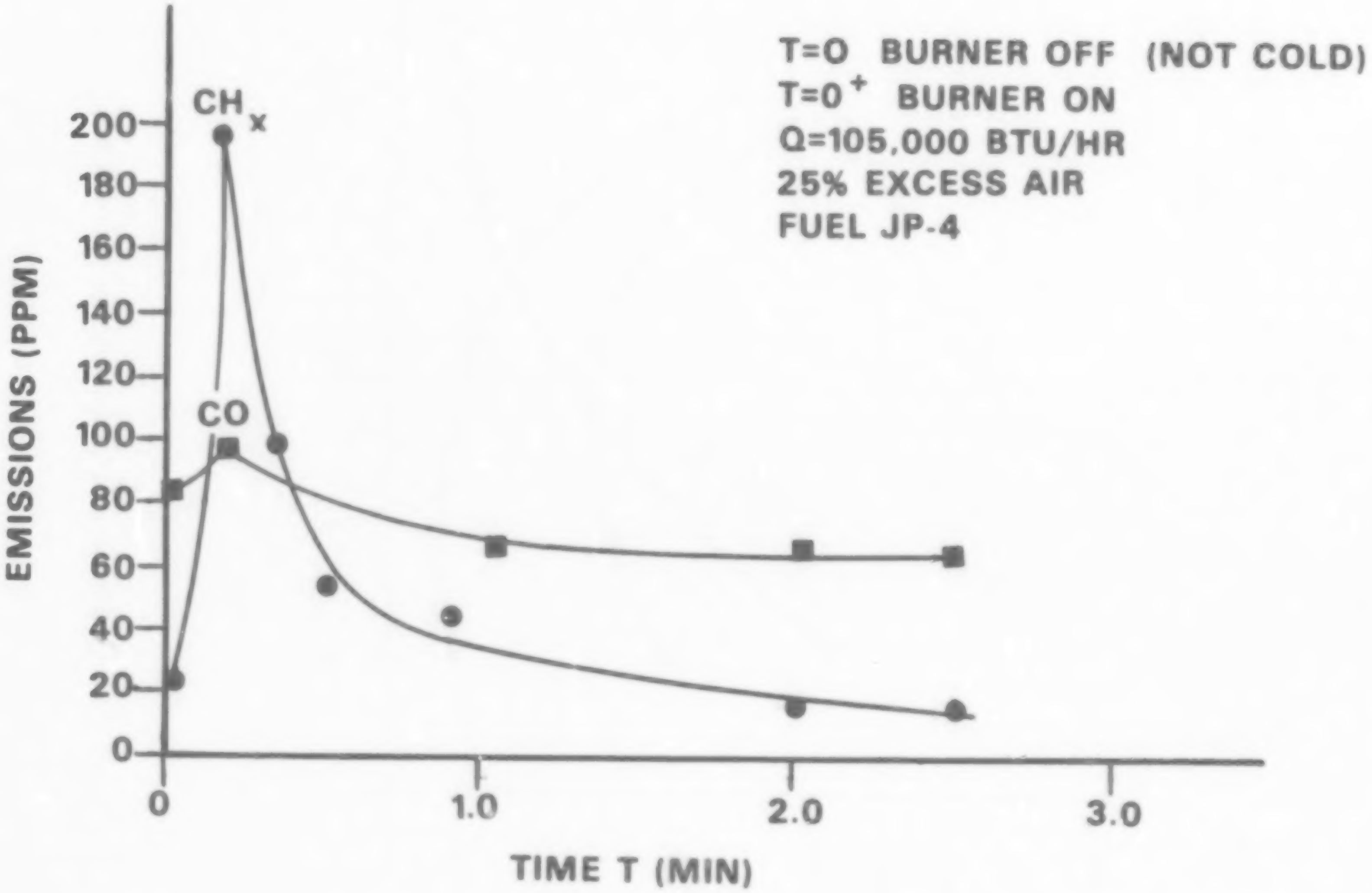


Figure 4.5.6 Burner On Test, 105,000 Btu/hr

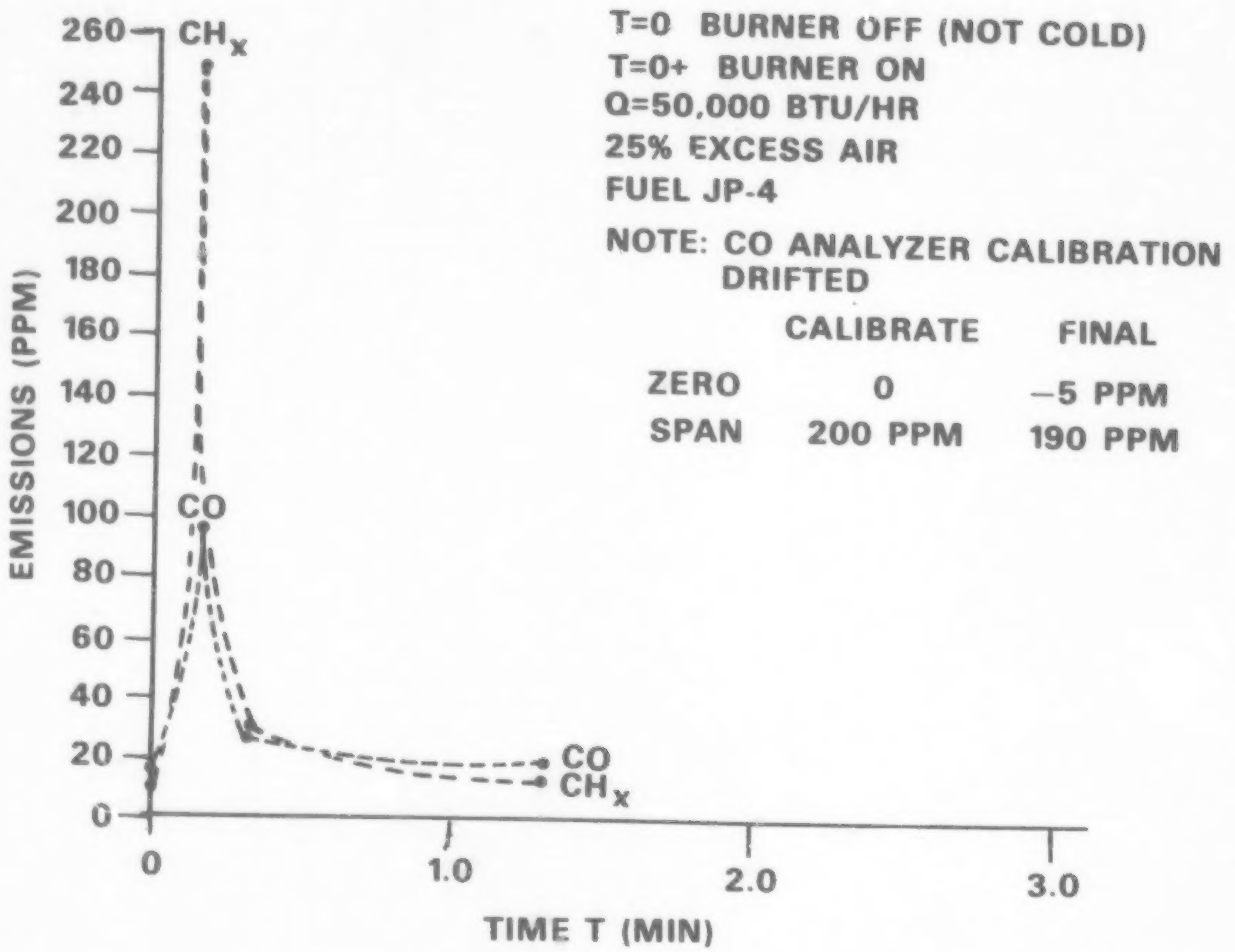


Figure 4. 5. 7 Burner-On Test, 50, 000 Btu/hr

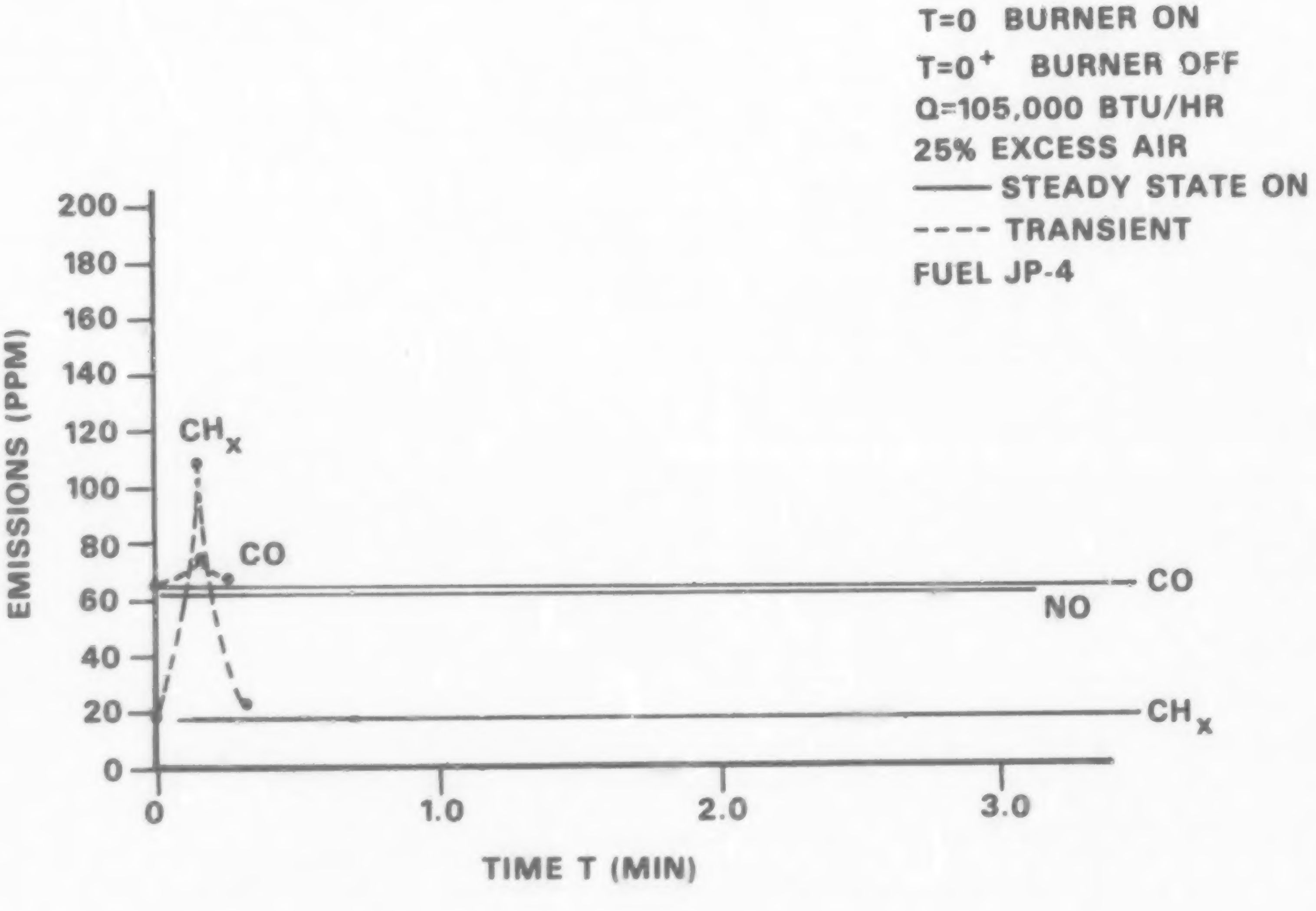


Figure 4.5.8 Burner-Off Test, 105,000 Btu/hr

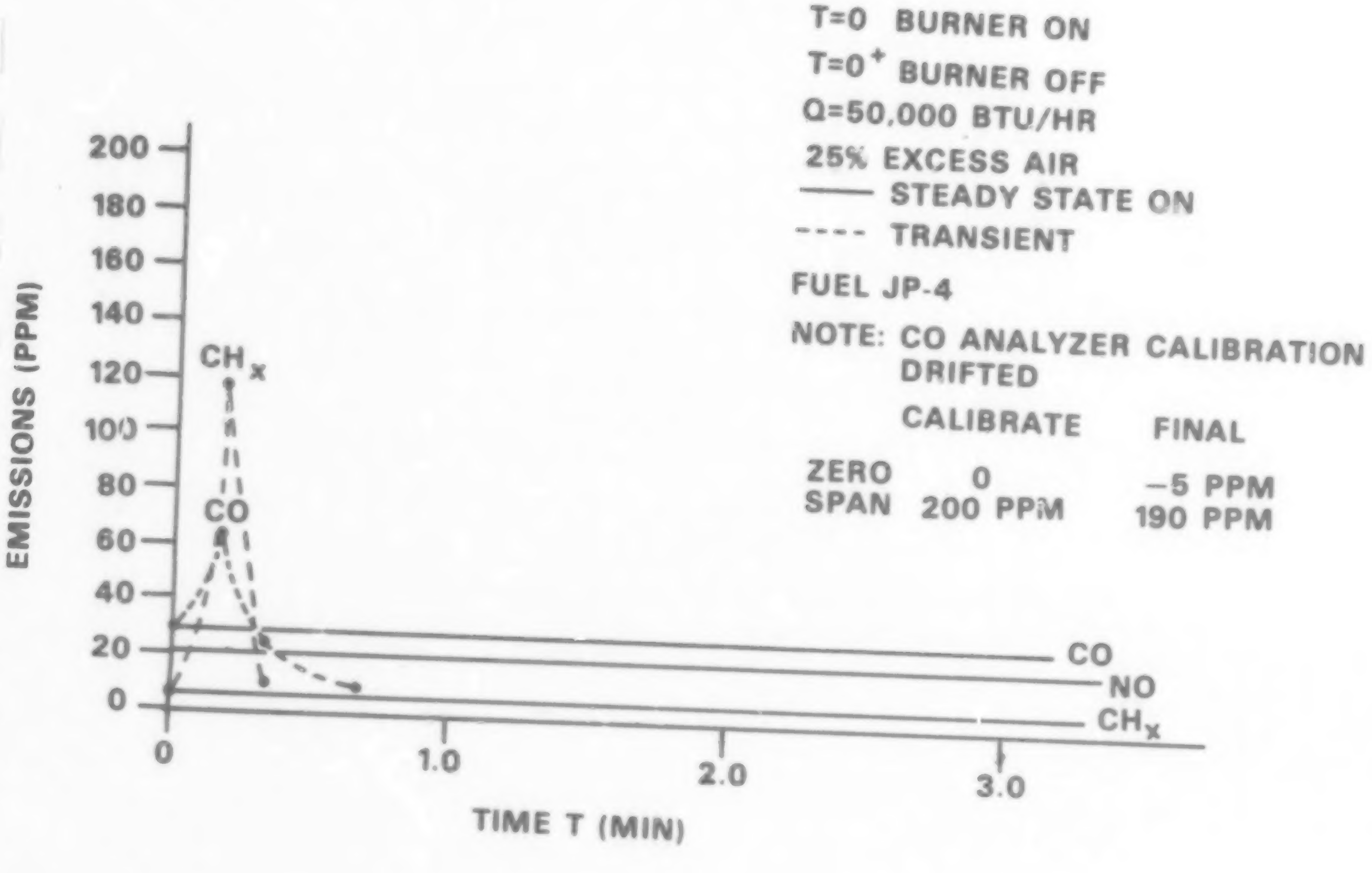
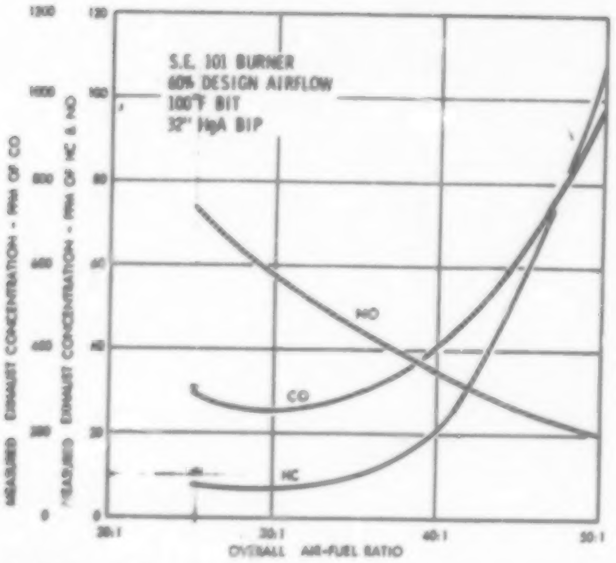
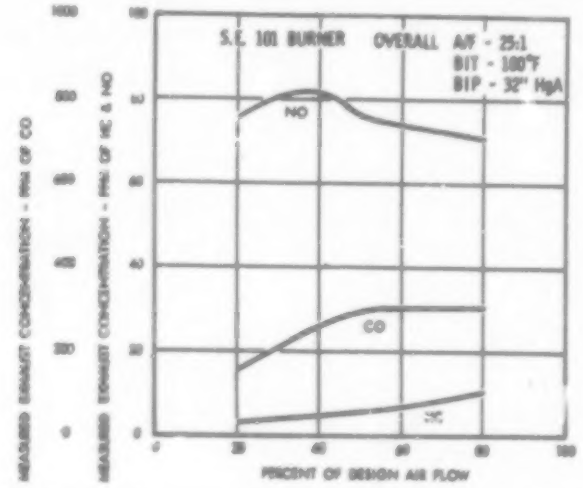


Figure 4.5.9 Burner-Off Test, 50,000 Btu/hr



a. Variation of measured exhaust emissions concentrations with overall air-fuel ratio. (9)



b. Variation of measured exhaust emissions concentrations with airflow. (9)

Figure 4. 5. 10

I 906

TABLE 4.5.4

TRANSIENT EMISSION DATA

FIRING RATE (BTU/HR)	EXCESS AIR (%)	CH_x (PPM)	CO (PPM)	NO_x (PPM)	ELAPSED TIME (MIN)
105,000	25%	6	60	-	0
50,000	25%	-	-	-	.5
50,000	25%	5	30	-	1.0
50,000	25%	-	25	-	1.5
50,000	25%	4.5	70	-	2.0
105,000	25%	40^1	180^1	-	2.5
105,000	25%	7	90	-	3.0
105,000	25%	5	-	-	3.5
50,000	25%	5	80	-	4.0
50,000	25%	4	25	-	4.5
50,000	25%	4.5	20	-	5.0
50,000	25%	-	-	-	5.5
105,000	25%	15^1	$1000+^1$	-	6.0
105,000	25%	-	-	-	6.5
105,000	25%	6	75	-	7.0
50,000	25%	-	75	-	7.5
50,000	25%	4.5	30	-	8.0
50,000	25%	4	15	-	9.0
				42^2	
				39^2	

NOTES:

1. Short duration peak ~ 10 sec.
2. Exhaust gas sample collected during the 9 minute run. Then two samples were drawn and NO analysis performed.

has low UHC and relatively low NO emission with a relatively high CO emission. The Marquardt burner has very low UHC and CO and relatively high NO emissions. The TECO burner is simultaneously low in all three contaminants at 25% excess air ratio. The results from all three independently developed burners confirm, in general, that very low emission levels will be obtained from a Rankine-cycle automotive propulsion system, provided a reasonable overall system efficiency is attained.

4.5.3 Some Considerations in Fuel Selection

The main considerations in selection of the fuel are:

a. Low ash and sulfur content.

b. Easy ignition and clean burning (volatility or low end of fuel affects ease of burning).

c. Safety (flash point of fuel).

d. Availability.

Short-term:	Available at local distribution for large Otto-Rankine ratio.
Long-term:	Available at local distribution for large Rankine-Otto ratio with minimum processing, considering other demands such as kerosene for aircraft and No. 2 fuel oil for heating.

In Tables 4.5.5 and 4.5.6, some of the fuel characteristics influencing these considerations are presented.(10)

TABLE 4.5.5

BROAD CONSIDERATIONS OF VARIOUS PETROLEUM FUELS

Product	Present Volume Availability	Retail Distribution		Performance		Approx. Bulk Price ¢/gal	Commentary
		Present	Future	Advantages	Disadvantages		
Propane	Limited	Limited		Clean Burning, Easy Ignition & Startup	Needs Pressure System, Safety, Low BTU Content	7.5[d]	Ideal from Standpoint of Emissions, Startup, Evaporation Loss
Gasoline	Ample	Wide Spread	?	Easy Ignition and Startup, Good Handling, Low Sulfur	Safety, Burner Deposits and Corrosion, Vaporization Loss	12.25[a]	Probably Unacceptable because of TEL and Halide Scavengers. Possibly will be Available Unleaded (?)
JP-4	Ample	Nil	Potentially Wide Spread	Easy Ignition and Startup, Good Handling, Low Sulfur, Reduced Vaporization Loss	Safety	--	Maximum Availability, Only Minor Processing Required in Manufacture
Jet A, No. 1 Burner (or Diesel), Kerosene	Limited	Limited		Easy Ignition and Startup, Good Handling, Low Sulfur, Safety, Little Vapor Loss	None	10.75[b]	Good Performance and Handling, Serious Problem of Future Availability. Could be solved by Processing if Market Demand Justifies.
No. 2 Burner (or Diesel)	Ample	Broad but Thin	Potentially Wide Spread	Higher BTU/gallon, Basically Same Advantages as No. 1	Cold-Flow at Extreme Temperatures, Smoke (?)	10.0[b]	Good Compromise between Performance, Safety, Availability, Cost, and Distribution Factors
No. 3 GT (Turbine)	Limited	Nil	Unlikely	High BTU/gallon/$	Higher Emissions, Handling (?), Ignitability, etc.		Interesting "Economy" Fuel--Essentially a low-ash resid
Nos. 4,5,6 (Residual)	Ample	Nil	Unlikely	High BTU/gallon/$	High Emissions, Deposits and Corrosion	5-6[c]	Probably Unacceptable on Performance Basis

[a] Oil and Gas Journal of October 6, 1969. "--quotations are realistic spot prices for refined products moving interstate on Wednesday each week. They will differ from refiners' prices for branded products. They should not be considered as postings." This is for 94 octane regular. Chicago (vs. 23.08 at the pump ex tax, 33.74 including federal, state and local taxes on national average).

[b] Same source, Chicago area

[c] Same source, New York Harbor, No. 6 in barges

[d] Platt's Oilgram, Producers Propane Prices in tank cars, transport truck, or pipeline input. New York Harbor, effective Oct. 1, 1969

BLM/kjk
10/20/69

TABLE 4. 5. 6

NOMINAL PHYSICAL PROPERTIES ACROSS THE FUELS SPECTRUM

Product	Distillation, °F					Gravity		Viscosity, cs		Sulfur,%	Ash,%	%H	BTU/Gal, Total
	Initial	10%	50%	90%	End Point	°API	Specific	-40°F	125°F				
Propane	--	--	--	--	-44	145	0.510	--		0.001	--	18.2	90,000
Gasoline	90	120	210	330	410	62	.731	1.5	0.5	0.03	0.1[a]	14	123,700
JP-4 Jet Fuel (or Type B)	140	210	290	390	460	54	.763	3.0	0.8	0.05	--	15	128,000
Type A Jet Fuel	330	370	410	480	520	43	.811	11	1.2	0.07	--	14	134,100
No. 1 Burner Fuel (or Diesel)	340	380	430	490	530	42	.816	15	1.3	0.10	--	14	134,700
No. 2 Burner Fuel (or Diesel)	370	430	510	590	630	34.5	.853	40[b]	2.1	0.30	--	13	139,100
No. 3-GT (Turbine)	--	--	--	--	--			>215[c]	>4[c]	--	<.03	--	~145,000
No. 4 Residual	--	--	--	--	--	19	.940	4,300[b]	10	1.0	.01	12	148,800
No. 5 (light) Residual	--	--	--	--	--	17	.953	22,000[b]	25	1.7	.02	12	150,000
No. 5 (heavy) Residual	--	--	--	--	--	14	.973	--	55	1.6	.04	11	152,000
No. 6 Residual	--	--	--	--	--	10	1.000	--	330	1.8	.06	11	154,600

[a] At 2.5 gm/gal Lead

[b] Minimum. Wax separation in most product occurs above this temperature and viscosities could be much higher.

[c] Corresponding to 45 SUS @ 100°F minimum in proposed ASTM specifications.

4.6 BOILER DESIGN

The reference cycle boiler heat transfer rate is 1.58×10^6 Btu/hr. The boiler must have some excess capacity for control purposes, however, and the maximum boiler heat transfer rate has been selected as 1.70×10^6 Btu/hr. With a boiler overall efficiency of 82.5%, this rate corresponds to a burning rate requirement of 2.06×10^6 Btu/hr (HHV). The boiler design goals were:

a. Positive elimination of hot spots on the organic side.

b. Boiler efficiency of at least 82.5% without air pre-heat and in compact boiler with low pressure drops.

c. Low material cost and easy construction.

To positively eliminate hot spots, a double-tube boiler construction is used with water sealed in the annular space between the tubes under its own vapor pressure. Heat transmission through the narrow (1/16 inch average thickness) water jacket occurs by boiling on the outside tube inner surface and condensing on the inside tube outer surface without any net circulation of water. Since the water is under its own vapor pressure, the temperature which the organic sees can never exceed the saturation temperature corresponding to the pressure in the water jacket, thereby positively prohibiting hot spots.

In Figures 4.6.1 and 4.6.2, cross sectional views through the combined burner-boiler are presented. In Figure 4.6.3, a top view of the tube bundle is illustrated. With reference to these figures, the combustion gases at about 3300°F flow downward through a duct into the central plenum formed by the boiler tube bundle; a screen distributor (ceramic-coated stainless steel) is used to provide a uniform flow through the tube bundle. The combustion gases then flow radially outward through the three stages of the tube bundle and are directly exhausted on leaving the tube bundle.

The organic and combustion gas flow paths are illustrated in Figure 4.6.4. The incoming organic first flows through the outer tube bundle (Stage 1) from which the combustion gases are exhausted; this provides the lowest organic temperatures in the boiler at the combustion gas outlet and a high boiler efficiency. It is important that an extremely compact heat exchange surface with high heat transfer rate per unit volume be used in this stage to maximize the boiler efficiency with an acceptably low pressure drop on the combustion side. In Figure 4.6.5, a comparison is given of several compact surfaces with the ball matrix in terms of the heat transfer rate per unit volume versus the power requirement per unit volume; the superiority of the ball matrix on this basis is readily apparent. In addition, the matrix can be easily fabricated between the round tubes required because of the high water jacket pressure.

Leaving Stage 1, the organic next flows into stage 2 through which the combustion gases first flow. Because of the high gas temperature and finned surface on the combustion side, coupled with a high heat transfer coefficient on the organic side, a very high heat transfer rate can be obtained in the second stage. The organic then flows through the superheater coil, or stage 3. This stage is a bare tube coil, since the controlling thermal resistance is on the organic side and an extended heat transfer surface is not required on the outside of the tube.

A boiler computer program has been developed for the detailed heat transfer and pressure drop analysis of the boiler. In Table 4.6.1, the boiler reference design point characteristics are summarized

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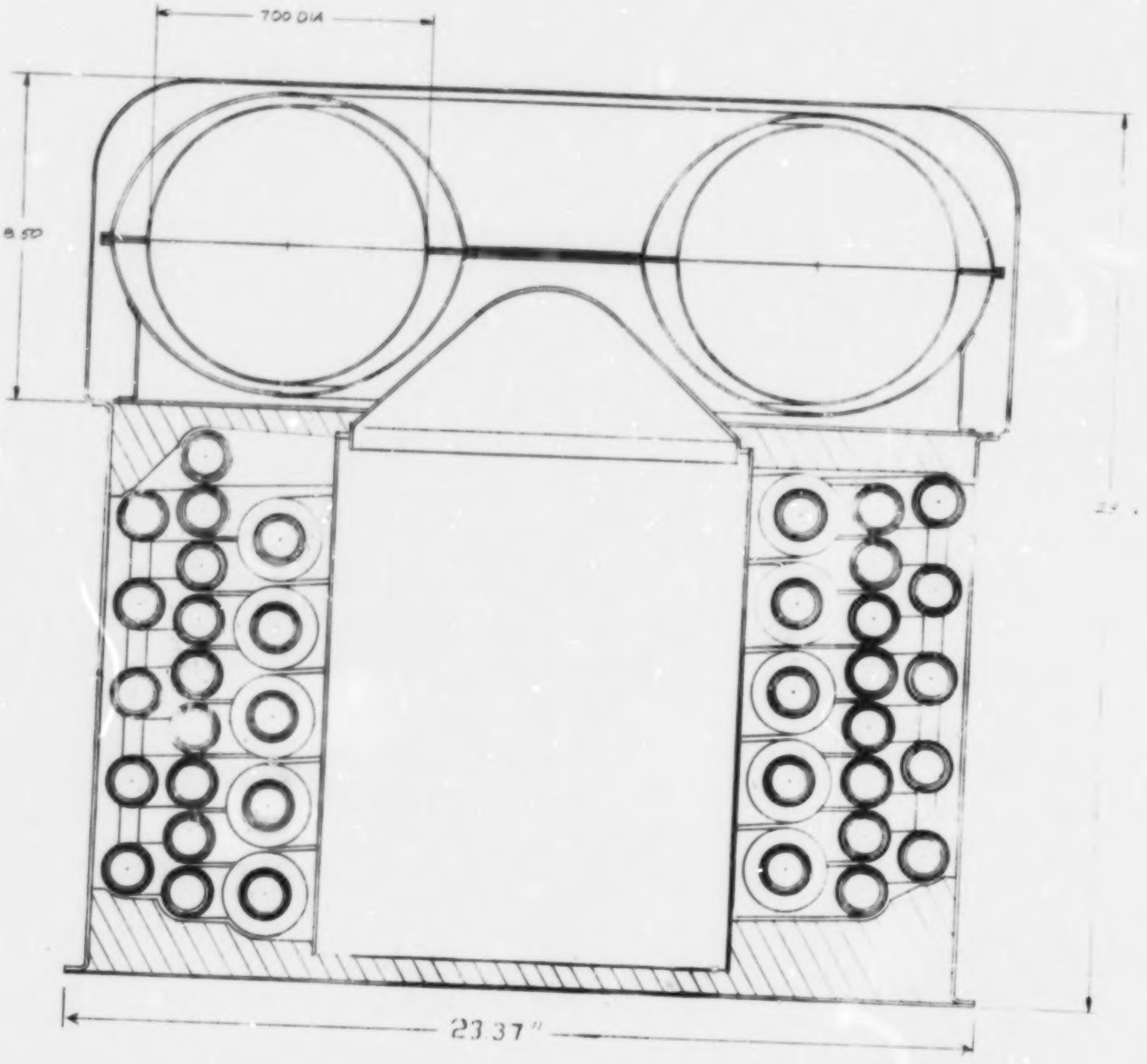

7.00 DIA
8.50
23.37"

Figure 4.6.1 Cross Section Through Burner-Boiler, Long Axis.

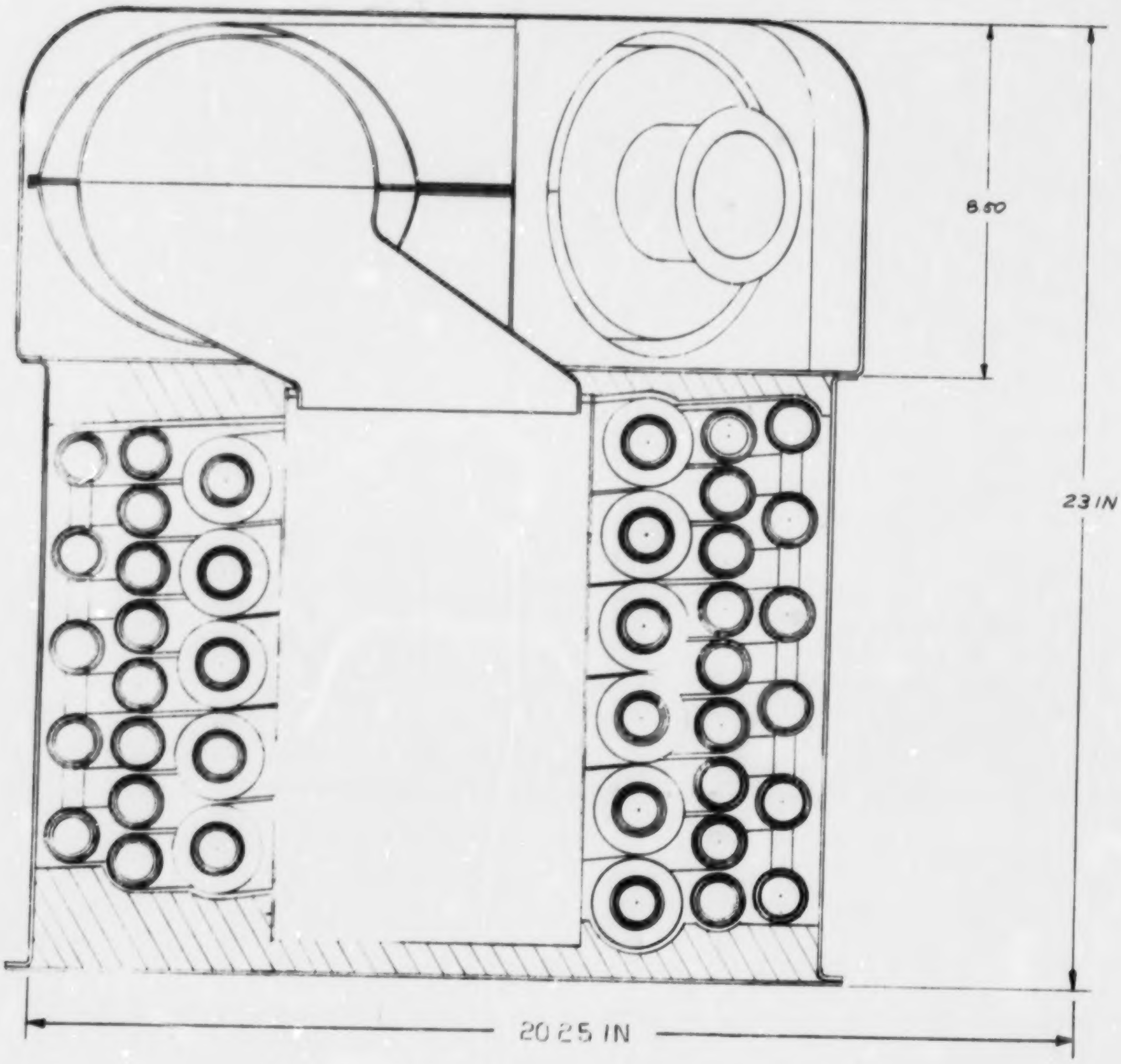

8.50
23 IN
20.25 IN

Figure 4.6.2 Cross Section Through Burner-Boiler, Short Axis.

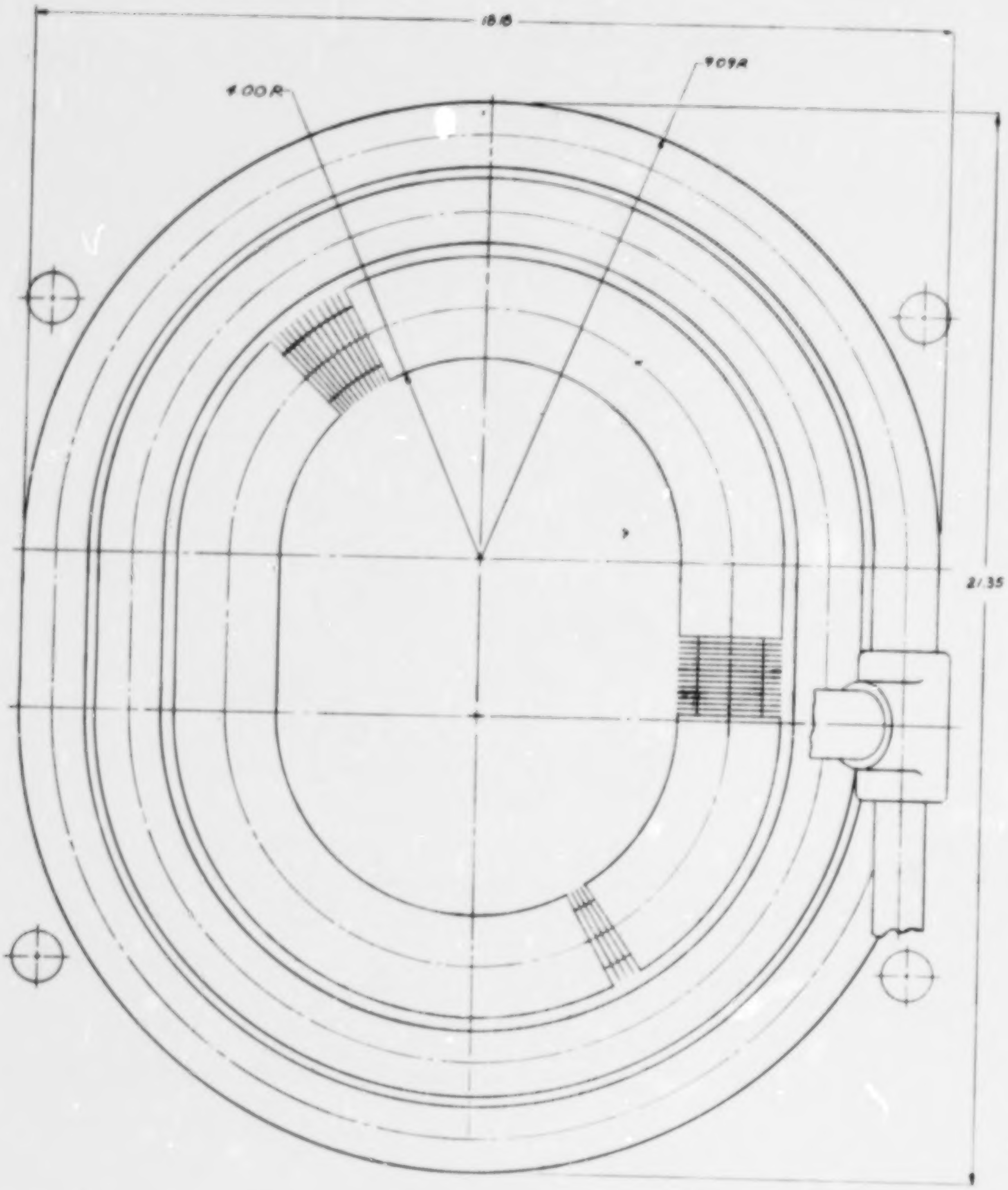

18.18
4.00R
9.09R
21.35

Figure 4.6.3 Top View of Boiler Tube Bundle.

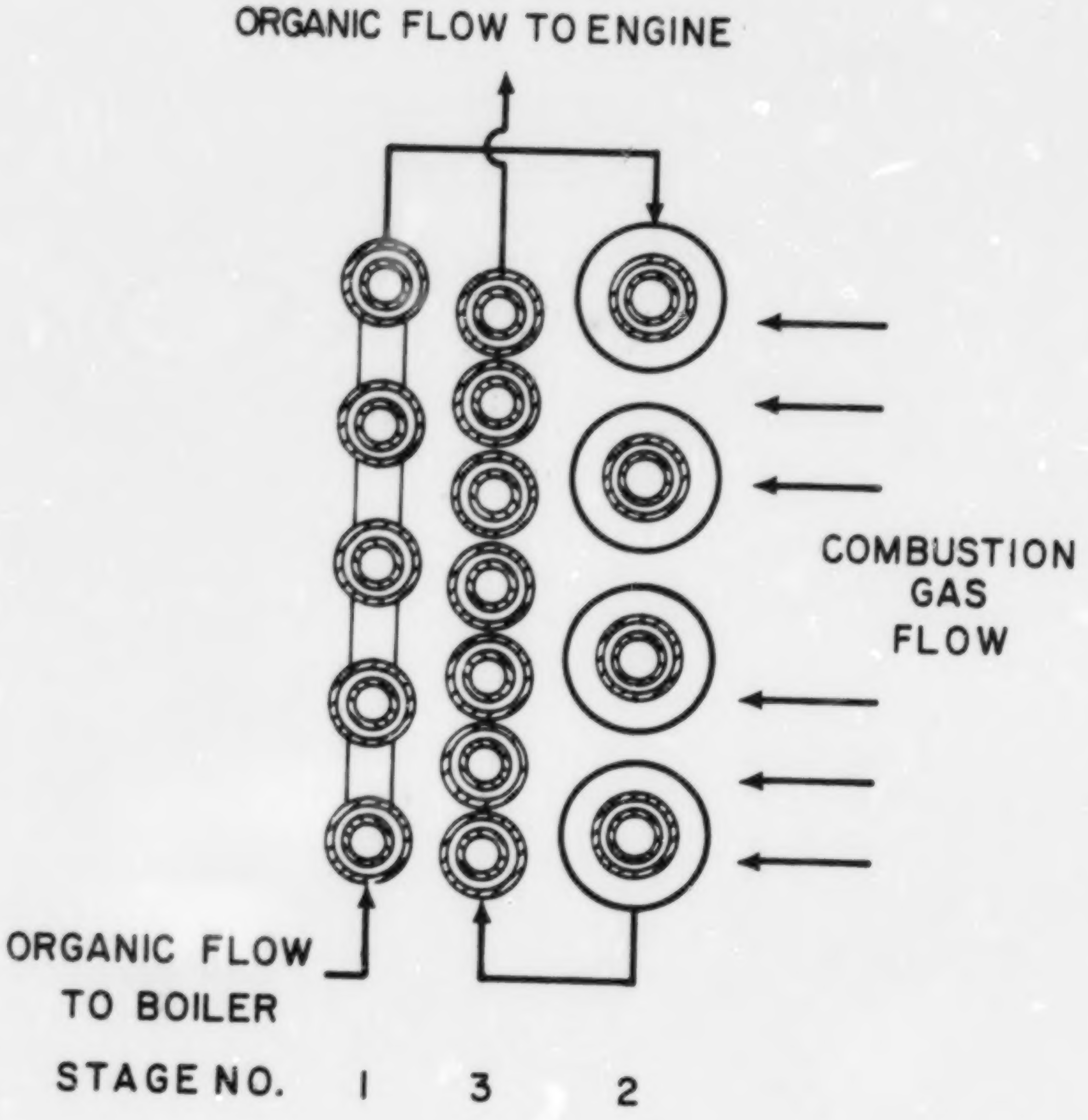
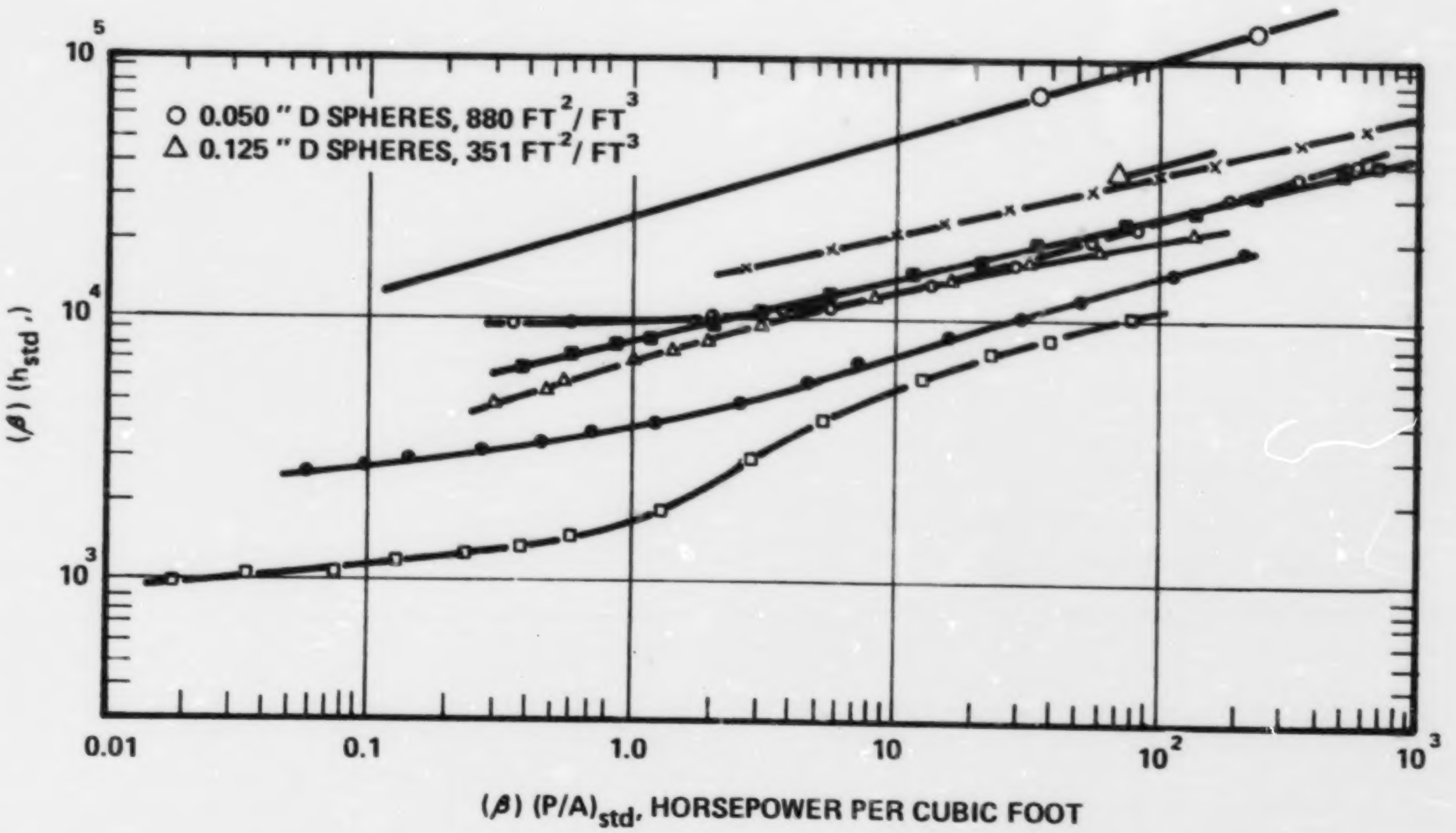

ORGANIC FLOW TO ENGINE
COMBUSTION GAS FLOW
ORGANIC FLOW TO BOILER
STAGE NO. 1 3 2

Figure 4. 6. 4. Organic Flow Path Through Boiler Tube Bundle



KEY	TYPE OF SURFACE	CODE NUMBER	β FT²/ FT³
×	RUFFLED FINS	17.8 - 3/8 R	514
△	IN LINE PIN FINS	AP-2	244
■	LOUVERED PLATE FINS	3/8 - 11.1	367
○	PLAIN PLATE FINS	19.86	561
□	INSIDE CIRCULAR TUBES	ST-1	208
●	FINNED FLAT TUBE	9.68 - 0.87	305

Figure 4.6.5 Comparison of Compact Exchanger Surfaces Illustrating High Efficiency of Ball Matrix.

TABLE 4.6.1

BOILER REFERENCE DESIGN POINT CHARACTERISTICS

Stage	Heat Transfer Rate Btu/hr	Combustion Gas Temperature °F		Tubing Length ft.	Pressure Design		Mass, Pounds					
		Entering	Leaving		Combustion Side in w.c.	Organic Side psi	Outer Tube Carbon Steel	Inner Tube Carbon Steel	Matrix Carbon Steel	External Fins Copper	Water	Total
Ball Matrix	359,000	1190	490	26.0	2.06	1.96	32.4	9.4	25.4	-	2.3	69.5
Finned	834,000	3330	1896	17.0	0.136	21.22	21.2	9.6	-	14.8	1.5	47.1
Superheater	383,000	1896	1190	35.0	0.288	23.32	43.6	12.7	-	-	3.2	59.5
Total	1,576,000	-	-	78.0	2.48	46.5	97.2	31.7	25.4	14.8	7.0	176.1

Tube Specifications

Inner Tube ID = 0.930 in.
OD = 1.000 in.

Outer Tube ID = 1.125 in.
OD = 1.315 in.

Material Carbon Steel

Extended Surface Specifications

(1) Matrix

Ball Size	3/32 in.
Matrix Thickness	0.5 in.
Matrix Height (Between Tubes)	0.935 in.
Ball Material	Carbon Steel

(2) Stage 2 External Radial

Fins/inch	10.0
Fin Thickness	0.012 in.
Fin Height	0.356 in.
Fin Material	Copper

(3) Stage 2 Internal Longitudinal

Number of Fins Around Tube	16
Fin Thickness	0.0312 in.
Fin Height	0.120 in.
Fin Material	Carbon Steel

and the physical specifications of the boiler heat transfer surfaces are given. The three boiler stages have a combined mass of 176 lbs; to this value must be added the following part weights to obtain the total burner-boiler weight:

Tube Bundle (combined three stages)	176 lbs
Expansion Tanks for Water Jackets	12 lbs
Thermal Insulation, Top and Bottom, and Top and Bottom Plates plus Exhaust Enclosure	15 lbs
Combustion Chamber	22 lbs
Air/Fuel Supply plus Ignition and Flame Detection System	43 lbs
Fitting and Supporting Hardware (estimated)	5 lbs
Total	273 lbs

In Figures 4.6.6 and 4.6.7, the organic, water and average wall temperature variations through the boiler are presented for the reference design point (7377 lb/hr throughput) and for 4000 lb/hr throughput, respectively. In Figure 4.6.8, the organic pressure input to the boiler for a constant boiler outlet pressure of 500 psia is presented as a function of throughput; at the reference design point, the organic pressure drop is 46.5 psi.

At the reference cycle condition, the boiler overall efficiency, based on the higher heating value of the fuel, is 82.7%, equivalent to a temperature of 490°F for the exhaust combustion products. In Figure 4.6.9, the variation of the boiler efficiency with organic throughput is presented. At low flows, the boiler efficiency approaches 86.0%. The boiler efficiency as well as organic pressure drop vary only slightly with the organic inlet temperature.

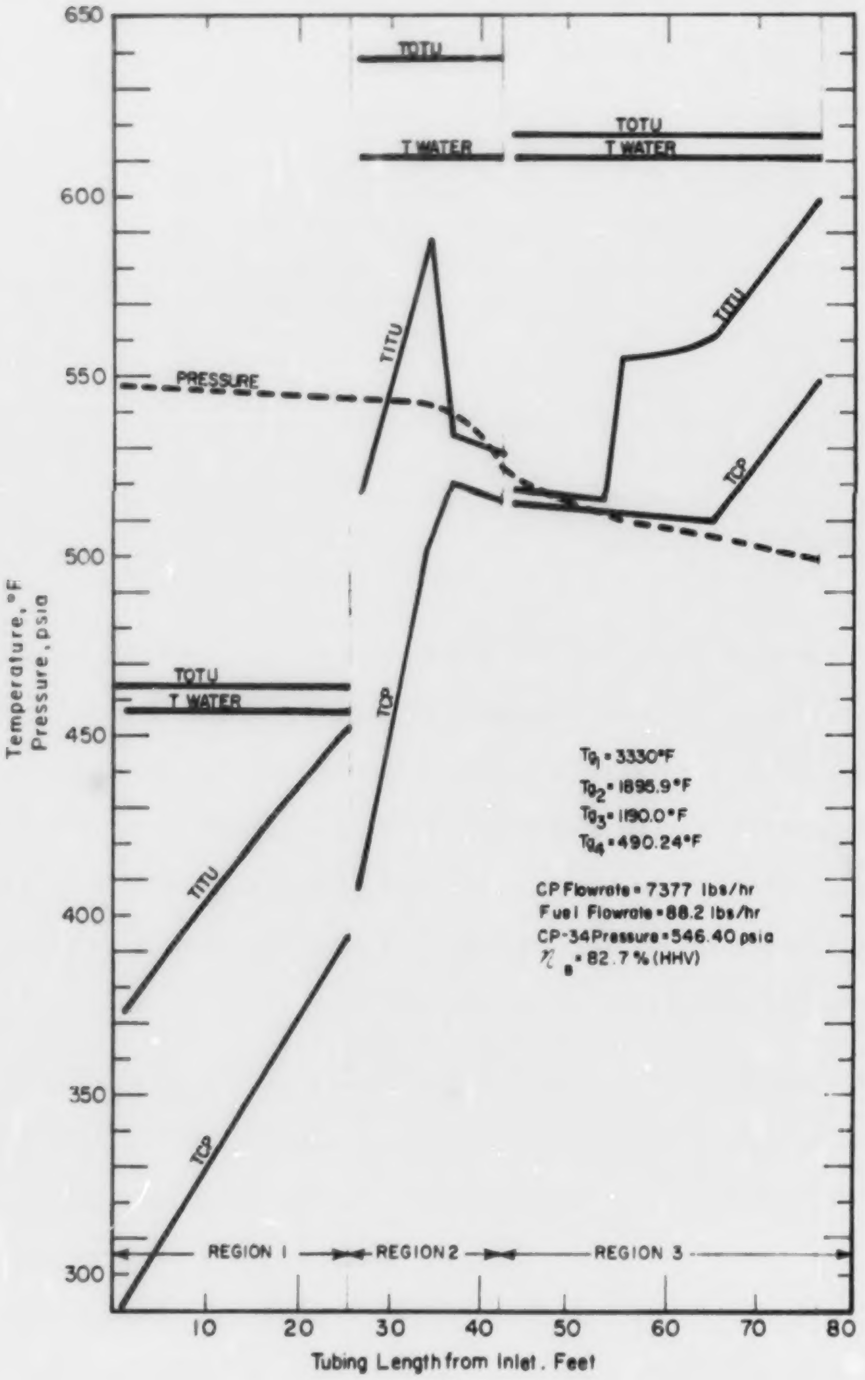


Figure 4.6.6. Design Point Boiler Temperature Profile

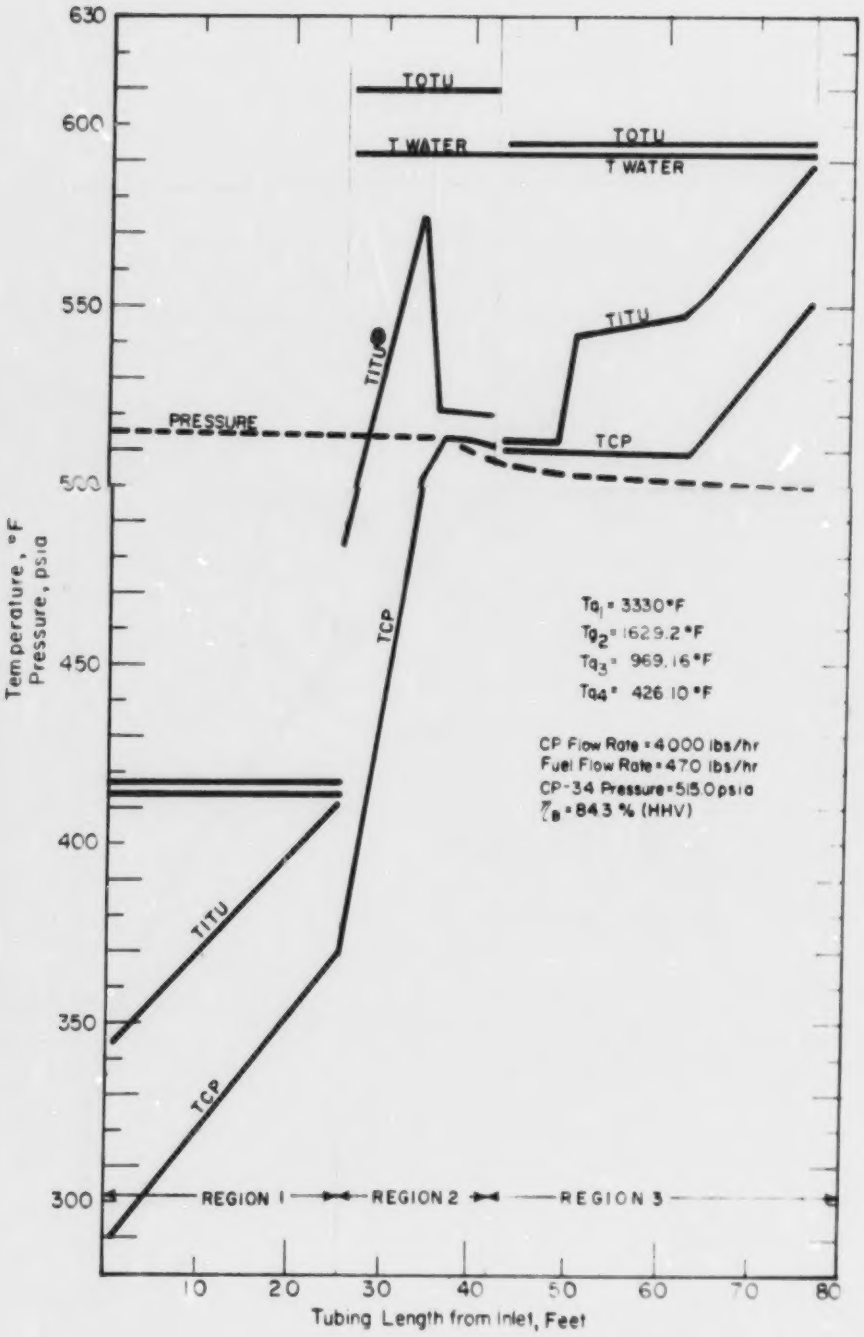


Figure 4.6.7. Medium Load Boiler Temperature Profil

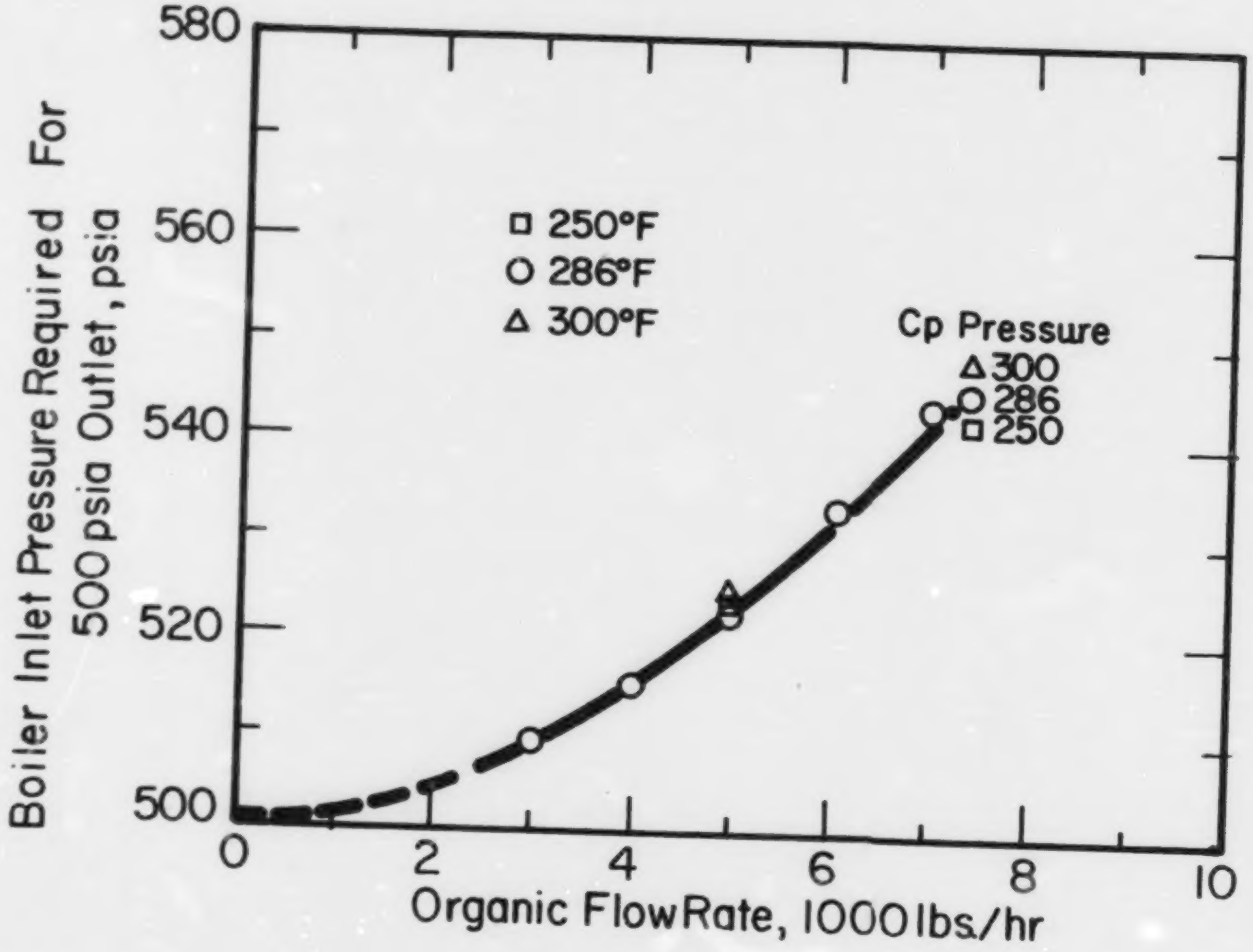


Figure 4. 6. 8 Organic Pressure Drop versus Throughput.

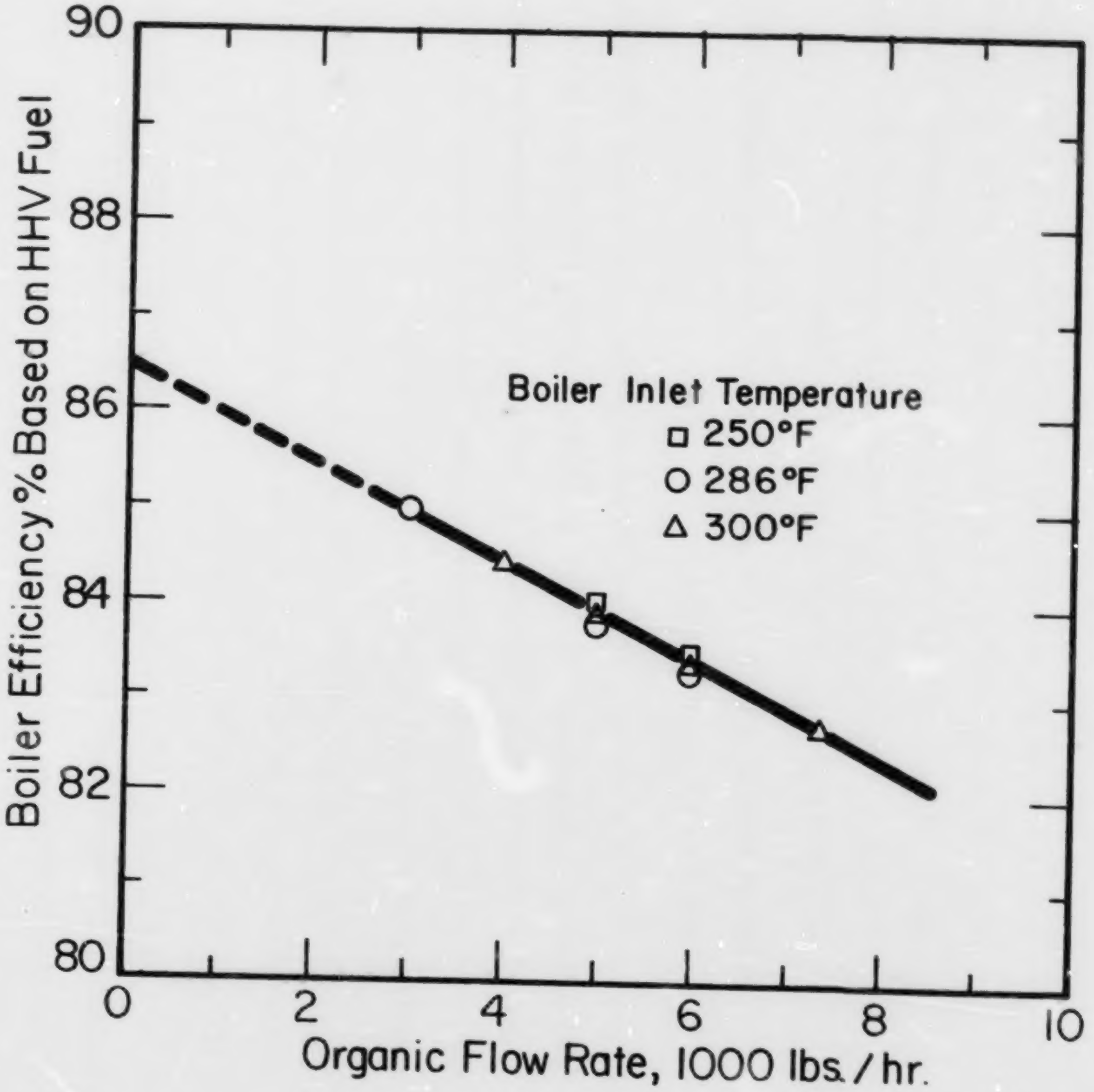


Figure 4. 6. 9 Boiler Efficiency (HHV) versus Throughput.

4.7 CONDENSER DESIGN

The condenser design in this study is based on the Ford radiator with louvered fins for the following reasons:

a. This fin type is amenable to mass production techniques at low cost as demonstrated by its use in automotive radiators.

b. The louvered fin heat transfer surface has acceptable pressure drop and fan power for a given frontal area and heat rejection rate. It does not necessarily represent the surface with minimum fan power, since manufacturing experience was given a high weight in selection of the fin type.

c. Heat transfer data on the finned surface were available from the Ford Motor Company.

In arriving at the design, the approach followed was to use the maximum frontal area available in the 1969 Ford Fairlane, with some sheet metal and frame modifications allowable at the front of the engine compartment. Use of the maximum frontal area minimizes the fan power required and results in a reasonable condenser configuration.

In Figure 4.7.1, the heat transfer and friction factor used in the design analysis is presented. These curves were derived from data supplied by the Ford Motor Company.

In Figure 4.7.2, the condenser design is illustrated. The condenser core measures 50 inches wide by 19.9 inches high by 3 inches deep; the basic core consists of copper fins, identical to those now used in the Ford radiator, and flattened carbon steel tubes extending through the depth of the condenser. The flattened tube has a greater thickness

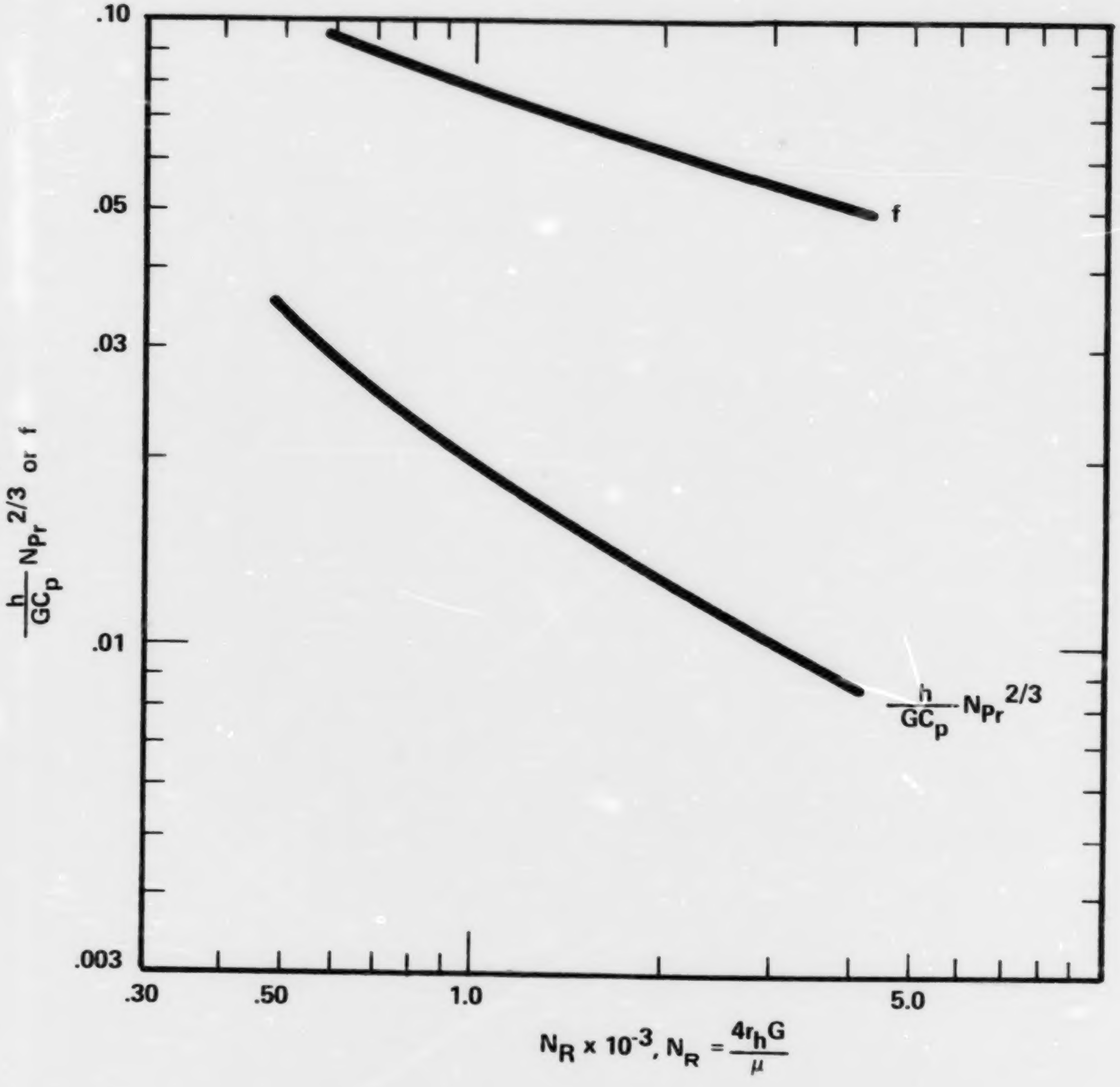


Figure 4. 7. 1 Heat Transfer Coefficient and Friction Factor versus Reynolds Number, Ford Louvered Radiator.

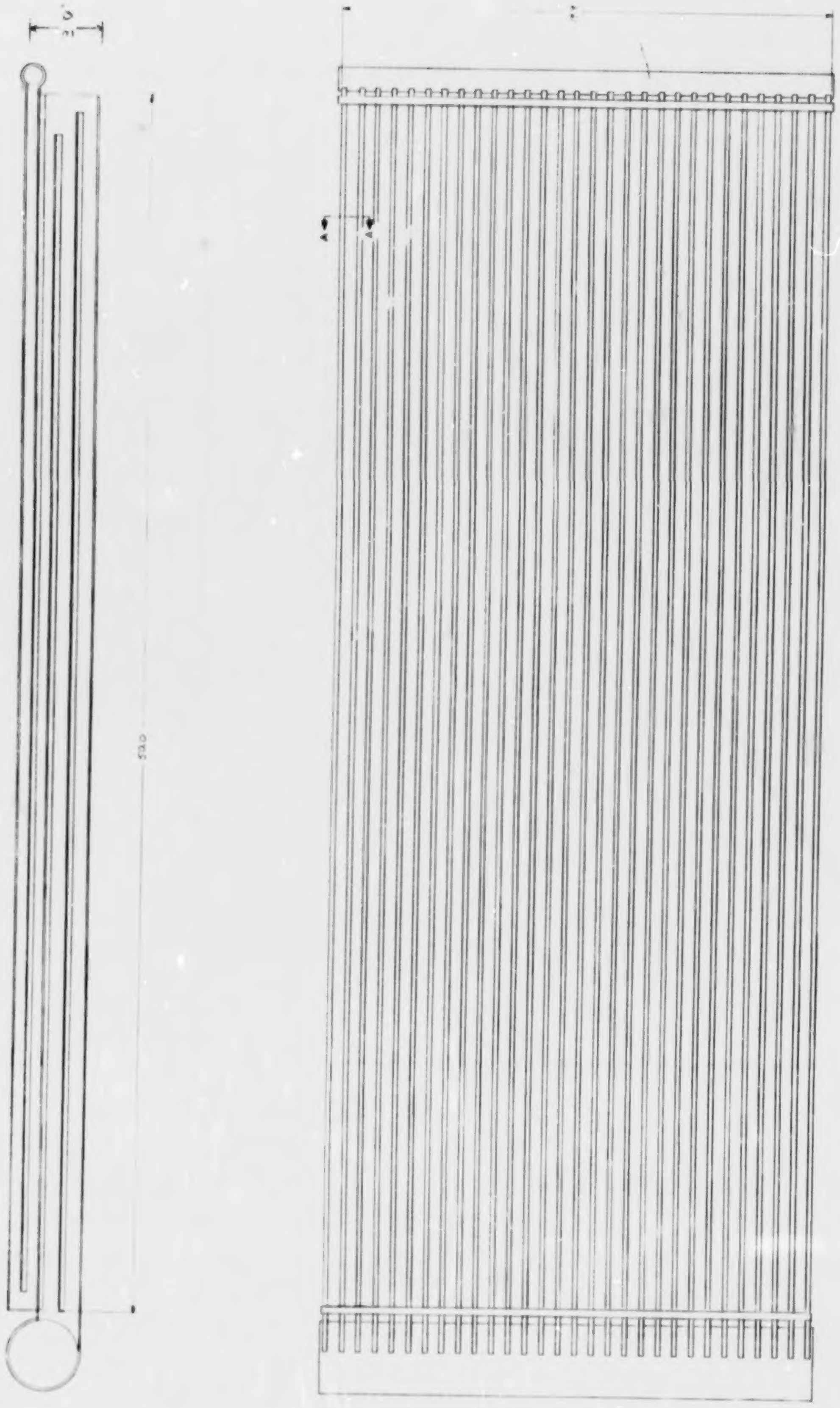


Figure 4.7.2a Condenser Design.

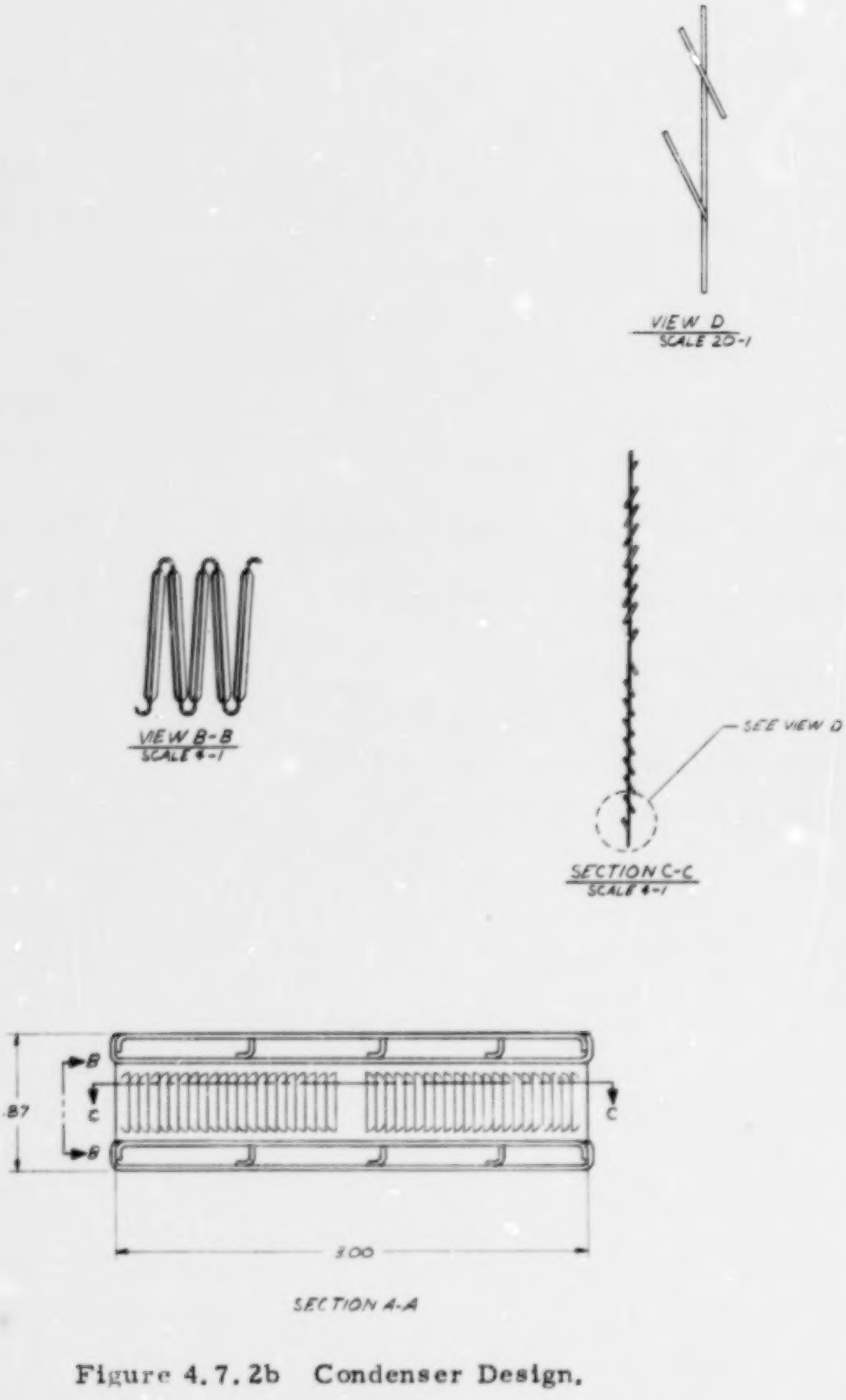

VIEW D
SCALE 20-1
VIEW B-B
SCALE 4-1
SEE VIEW D
SECTION C-C
SCALE 4-1
B
C
C
B
.87
3.00
SECTION A-A

Figure 4.7.2b Condenser Design.

TABLE 4.7.1

CONDENSER PHYSICAL CHARACTERISTICS

Flattened Tubes	
Number of tubes	30
Total Length	129 feet
Total Mass (Carbon Steel)	85.6 lbs
Fins	
Fins/inch	14
Fin thickness	0.0025 inch
Fin mass (copper)	25.5 lbs
Vapor Header Mass (Carbon Steel)	2.0 lbs
Liquid Header Mass (Carbon Steel)	1.0 lbs
Mounting Hardware	1.0 lbs
Total Condenser Mass (Carbon Steel plus Copper)	115 lbs

TABLE 4.7.2

CALCULATED CONDENSER PERFORMANCE AS A FUNCTION OF AIR FLOW RATE

```
     ØRGANIC RATE = 7377 LB/HR              ØRGANIC TEMPERATURE = 230 F                 AIR TEMPERATURE = 95 F

  P IN    AIR FLØW       HEAT ØUT  HIN-HØUT   T ØUT     X ØUT    P ØUT   AIR DRØP      HEAT DIST       LENGTH DIST       FAN PWR
  PSIA     LB/HR         MBTU/HR    BTU/LB      F                PSIA     IN H2Ø    VAP  CØND  LIQ   VAP  CØND  LIQ      HP

  27.0     90000          1.296     175.69    177.03    -.116    29.08     6.10     1.0  88.5  10.5   1.0  55.6  43.4    20.64
  27.0     80000          1.274     172.74    186.77    -.100    29.33     5.02     1.0  90.3   8.7   1.0  61.6  37.4    15.17
  27.0     70000          1.240     168.12    198.59    -.071    29.59     4.04     1.0  92.5   6.5   1.0  69.7  29.3    10.71
  27.0     60000          1.190     161.35    217.11    -.030    30.19     3.14     1.0  96.2   2.8   1.0  87.9  11.1     7.17
  27.0     50000          1.100     149.12    228.90     .048    30.20     2.33     1.1  98.9    .0   1.0  99.0    .0     4.46
*STØP* 0
```

than the tubing used in the radiator and has a greater wall thickness (0.030 inch versus 0.005 inch). Flow dividers are positioned in the flattened tube to provide three organic passes through the condenser and to serve as stays for the flattened tube walls. The thirty flattened tubes in the condenser are connected to common vapor and liquid circuits, comprising 30 parallel organic flow circuits.

Some condenser physical characteristics are summarized in Table 4.7.1, which shows that the total condenser mass when constructed of carbon steel tubing and copper fins is 115 lbs.

An alternative which is attractive from both a cost and a weight point of view is use of an all-aluminum condenser, using the same basic design with the following changes:

1. Increase wall thickness of headers.
2. Increase number of dividers in flattened tube.
3. Increase fin thickness from 0.0025 inch to 0.005 inch.

With these changes, the heat transfer performance should be approximately equivalent to that of the reference design and both the mass and manufacturing cost should be reduced significantly. The mass of the all-aluminum condenser would be as follows:

Tubes	30.8 lbs
Fins	15.5 lbs
Headers	3.0 lbs
Mounting Hardware	1.0 lb
Total	50.3 lbs

The Ford Motor Company has indicated the aluminum condenser should be considerably less expensive than the carbon steel and copper condenser.

A computer program for calculating the detailed condenser performance has been completed and incorporated into the generalized computer program for calculating the overall system performance. The condenser computer program calculates the heat transfer performance increment by increment, using the best techniques available for calculating the condensing coefficient and two-phase pressure drop through the organic side of the condenser In Table 4. 7. 2, a summary computer printout is illustrated for an inlet pressure of 27. 0 psia and inlet organic temperature of 230°F. Calculations were performed at different air flow rates; the table gives a summary of the condenser performance. The fan power is the shaft fan power, based on use of two Torrington A-2029-5 fans (20. 0 inches O. D and 3. 50 inch pitch). The "X out" is the quality of the organic effluent from the condenser; a negative quality refers to subcooled liquid It is planned to use the same heat exchanger as a combination desuperheater, condenser, and subcooler. The design criterion used is that 15% of the organic flow path length be used as the subcooler. By interpolation from Table 4. 7. 2, this occurs at an air flow rate of 61,000 lbs/hr, equal to 14,200 CFM of air flow at 95° F and 1 atmosphere. The total heat rejection for this condition is 1.20 x 10^6 Btu/hr.

4.8 REGENERATOR

The regenerator design is based on obtaining 90% effectiveness at the design point, with effectiveness defined as

$$E_{Reg} = \frac{h_i - h_o}{h_i - h_{sat}} ,$$

where h_i and h_o are the actual organic enthalpies in and out of the regenerator, and h_{sat} is the saturation enthalpy of the organic at the regenerator outlet pressure. Under part load conditions, the regenerator effectiveness will generally be greater than 90%.

As illustrated in Figure 4.8.1, the regenerator design is based on use of a porous ball matrix extended surface on the vapor side; four stages are used on the vapor side, since this provides performance close to a pure counterflow exchanger. Four parallel liquid passes are used to minimize the liquid side pressure drop. The regenerator design point characteristics are summarized in Table 4.8.1.

A computer program has been written and incorporated into the generalized system model. This program treats each stage separately in the calculation for increased accuracy.

The regenerator, as described in Section 5, is packaged on top of the engine. While not shown in detail, flanges on the exhaust ports will be cast in the engine block. The regenerator will be mounted directly on the engine by use of mating flanges on the regenerator vapor inlet. The vapor then flows upward through the ball matrix. This arrangement permits the regenerator to function as an effective oil separator, removing a major fraction of the engine lubricant blowby back to the crankcase through a drain line. This drain line

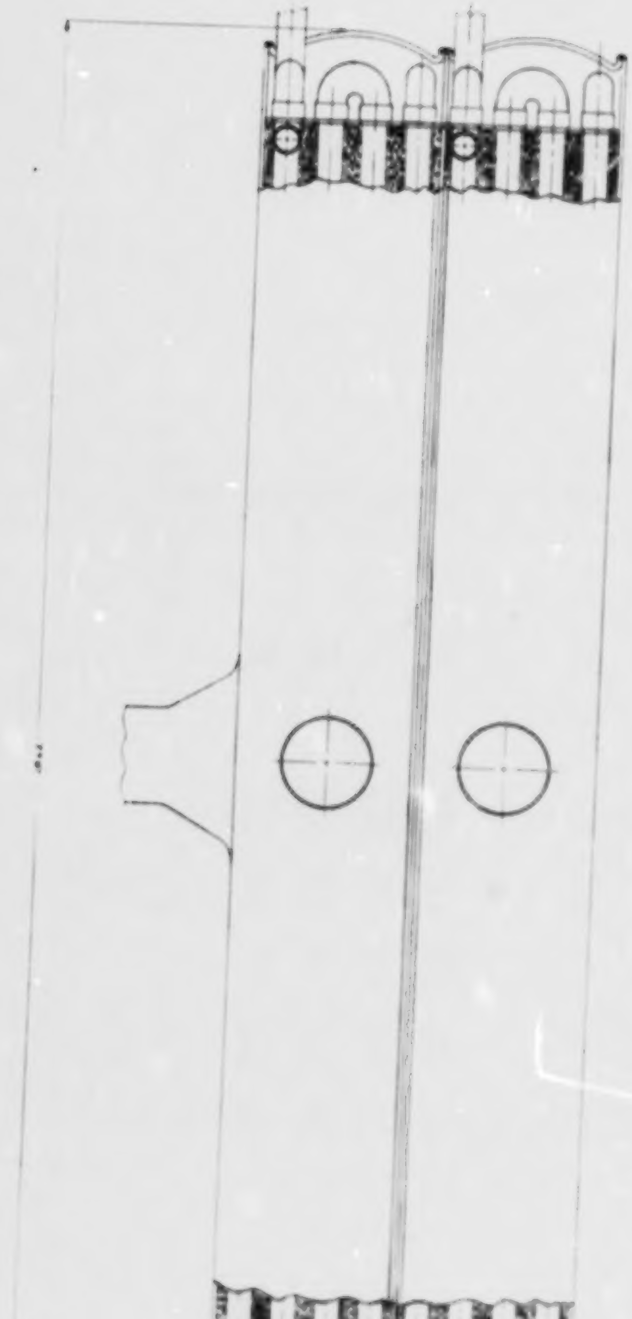


Figure 4.8.1a Regenerator Design

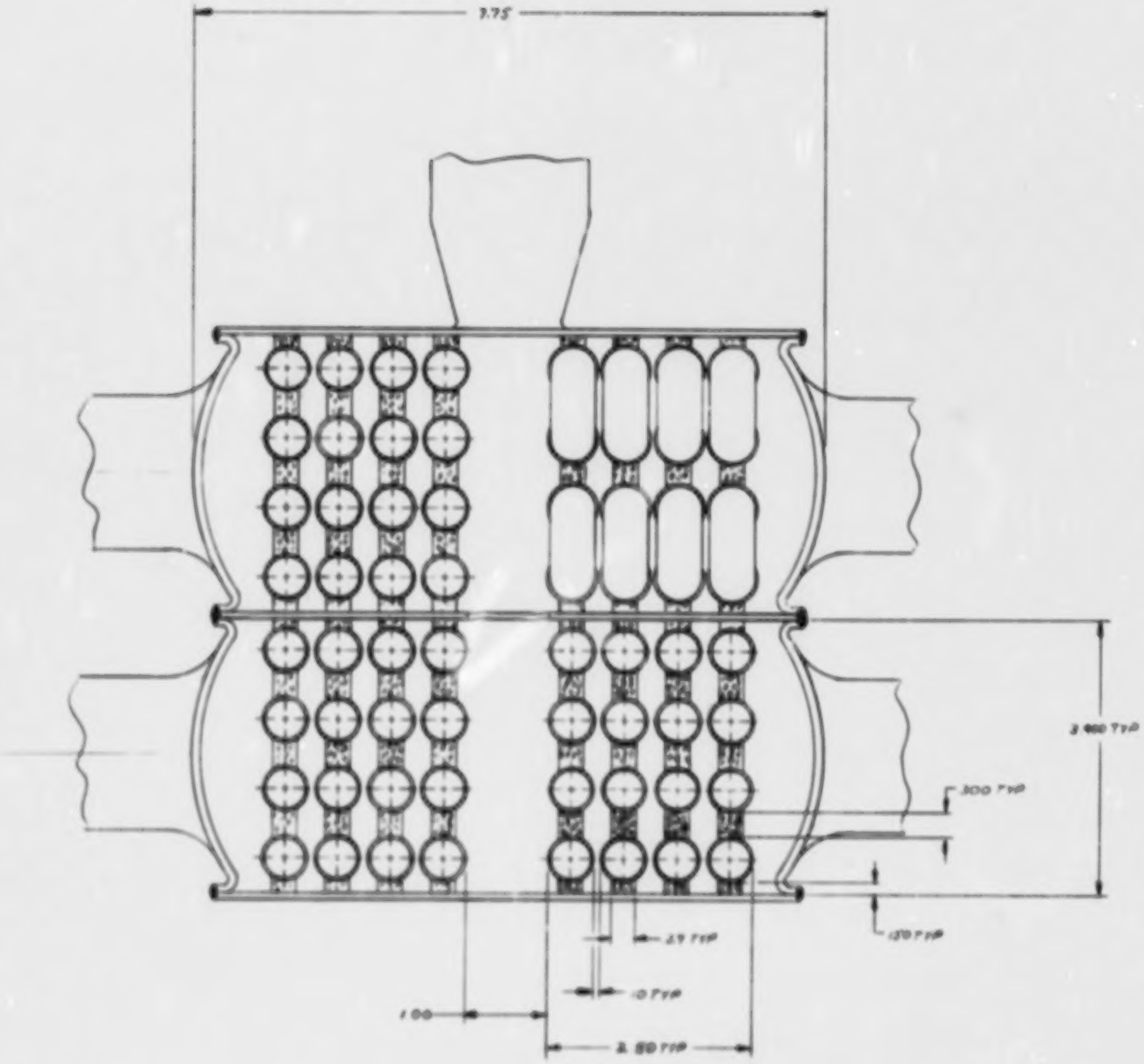

7.75
3.400 TYP
.300 TYP
.150 TYP
.27 TYP
.10 TYP
1.00
2.50 TYP

Figure 4, 8, 1b Regenerator Design.

TABLE 4.8.1

REGENERATOR DESIGN POINT CHARACTERISTICS

Heat Transfer Rate	249,000 Btu/hr
Effectiveness	90%
Vapor Temperatures	
Inlet	348.1°F
Outlet	230°F
Liquid Temperatures	
Inlet	199.0°F
Outlet	285.3°F
Vapor Pressure	
Inlet	25.0 psia
Pressure Drop	2.2 psia
Liquid Pressure	
Inlet	546 psia
Pressure Drop	3.9 psia
Number of Stages	4
Number of Parallel Liquid Passes	4
Tubing (Carbon Steel)	
Total Length	133-1/3 ft.
OD	0.550 inch
ID	0.500 inch
Weight	12.5 lbs.

TABLE 4.8.1 (continued)

Matrix (1/4 Copper Balls, 3/4 Aluminum Balls)	
Ball Diameter	1/16 inch
Matrix Height Between Tubes	0.303 inch
Matrix Thickness	0.29 inch
Weight, Carbon Steel Balls	17.9 lbs
Weight, Copper Balls	6.0 lbs
Total Matrix Weight	23.9 lbs
Shell	
Thickness	1/16 inch
Weight	16.0 lbs
Total Regenerator Weight (with 1.6 lb allowance for fittings, supports, etc.)	54.0 lbs

is also used as the crankcase vent line. Use of the regenerator for this purpose eliminates the requirement for a separate oil separator.

4.9 ROTARY SHAFT SEAL AND STATIC SEALS

4.9.1 Rotary Shaft Seal

Slight leakage of the working fluid past the piston rings into the crankcase (blowby) is inevitable. Thus, the crankcase must be considered part of the system volume within which the working fluid must be confined. A shaft seal must be provided where the crankshaft passes through the crankcase wall, since loss of working fluid and leakage of air into the system are both unacceptable. The vapor space in the crankcase is vented to the regenerator, making the crankcase pressure essentially equal to the pressure in the regenerator. During normal operation with thiophene, this pressure is higher than atmospheric pressure, and the crankshaft seal must prevent the loss of working fluid. During shutdown, the condenser pressure corresponds to the vapor pressure of the working fluid at the prevailing ambient temperature; since this pressure will generally be less than atmospheric, the crankshaft seal must prevent the leakage of air into the crankcase.

Since the power plant is shut down for most of its lifetime, leakage of air into the system represents the most serious difficulty. Air leakage into the system has two detrimental effects. First, the presence of oxygen accelerates the thermal decomposition of the organic material. Also, the noncondensible gases (both the air leaking in and the gases produced by thermal decomposition) collect in the condenser, degrading its heat rejection capability and reducing the overall system performance. Air leakage into the system must therefore be limited to extremely low levels.

Shaft seals which permit only very low fluid leakage in one direction (i. e., either into or out of the crankcase) are readily available. However, where leakage in both directions must be minimized due to a reversal of the pressure force, the double seal geometry shown in Figure 4. 9. 1 is used. The pressure of the buffer fluid is sufficient to ensure that both of the rotating shaft seals function as unidirectional seals. These unidirectional seals could be either lip seals or mechanical face seals. The choice between lip seals and face seals for the individual rotating seals shown in Figure 4. 9. 1 is dictated by the requirements of the application. The inboard seal is in a thiophene environment, wherein Viton, the only thiophene compatible elastomer, swells as much as 30%. A lip seal is not suitable here, since a long life, low leakage lip seal is not possible when the elastomer swells appreciably. Thus, a mechanical face seal is selected for the inboard seal.

The pressure of the buffer fluid in the seal cavity must be kept above the crankcase pressure of approximately 25 psia. A buffer fluid pressure of 30-35 psia is suitable. Therefore, the pressure difference across the outboard seal is 15-20 psi. Although lip seals are capable of pressure differentials of this magnitude, they do not provide long life at these relatively high pressure differentials. Thus, the outboard seal should also be a mechanical face seal for good performance of the system.

Figure 4. 9. 2 shows the recommended shaft seal design. The sketch shows the use of a single rotating seal ring which minimizes the axial length of the seal (approximately 2. 5 inches, as shown in

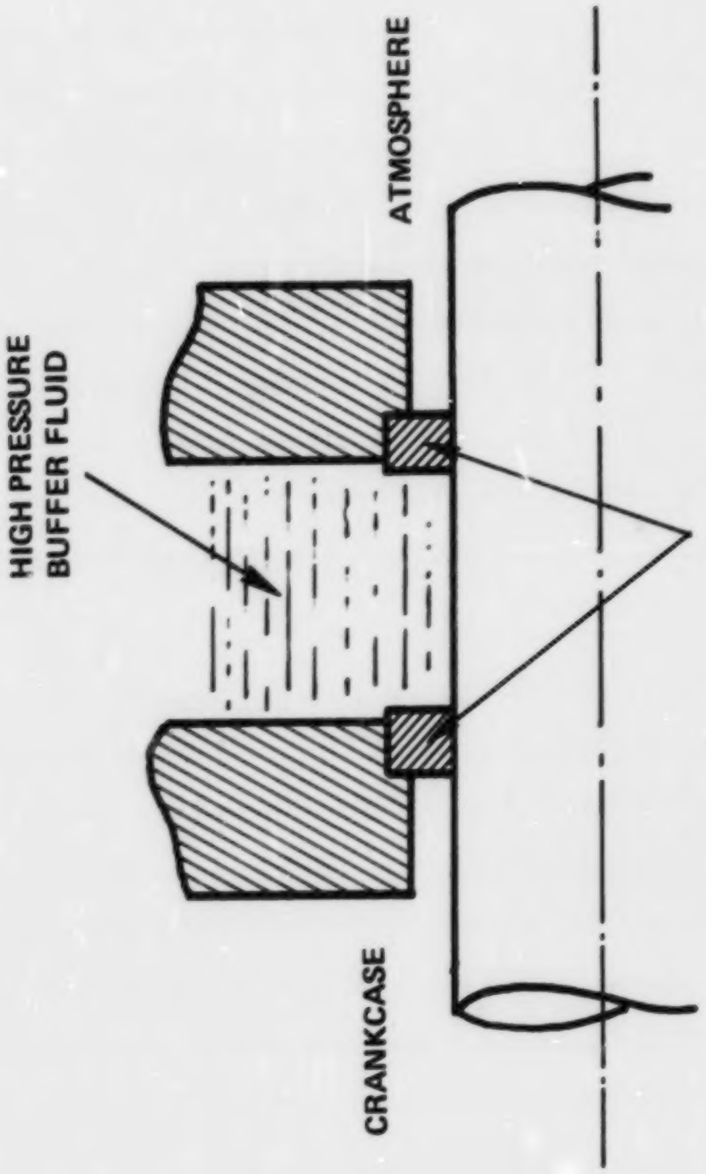

HIGH PRESSURE
BUFFER FLUID
ATMOSPHERE
ROTATING SHAFT SEALS
CRANKCASE

Figure 4.9.1 Double Shaft Seal Concept.

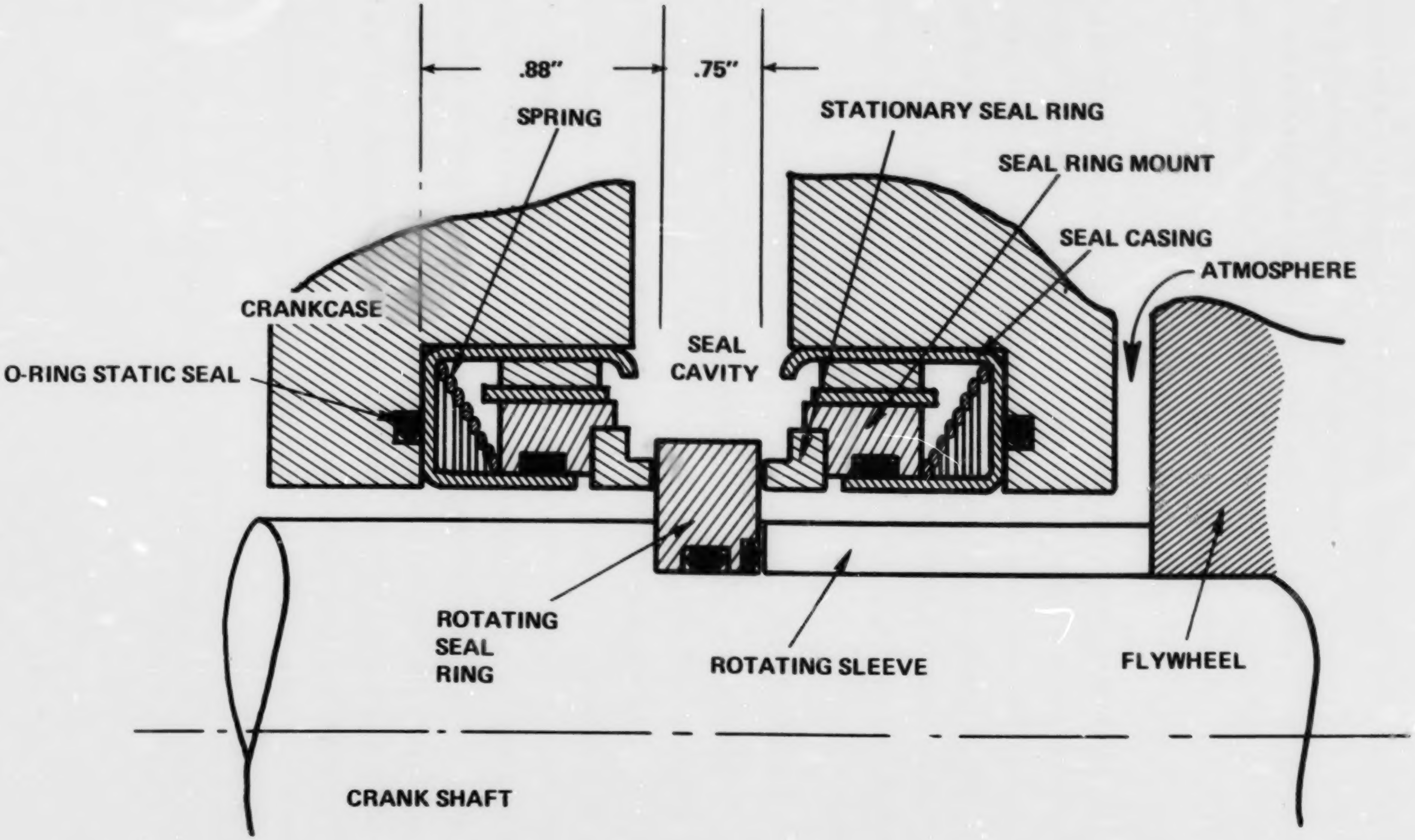

.88"
.75"
SPRING
STATIONARY SEAL RING
SEAL RING MOUNT
SEAL CASING
ATMOSPHERE
CRANKCASE
O-RING STATIC SEAL
SEAL CAVITY
ROTATING SEAL RING
ROTATING SLEEVE
FLYWHEEL
CRANK SHAFT

Figure 4. 9. 2 Shaft Seal Design.

Figure 4.9.2). Of course, there are several alternatives to this basic design. The retaining sleeve ensures correct axial positioning of the rotating seal ring by forcing the ring against the shoulder in the shaft. The five static O-ring seals shown in Figure 4.9.2 would use Viton O-rings. The stationary seal ring could be carbon and the rotating ring could be hardened stainless steel.

The buffer fluid pressure must be maintained at 30-35 psia during operation and at a minimum of 20-25 psia during shutdown. The simplest way to do this may be to keep the pressure constant at 30-35 psia using the system shown in Figure 4.9.3. The combination of the spring and bellofram keep the pressure of the buffer fluid constant by allowing enough volume displacement to compensate for leakage past the shaft seals.

4.9.2 Static Seals

Static seals, which are required at several locations in the system, must be vacuum tight. While O-rings could technically be used, machining of O-ring grooves is too expensive; use of O-rings is thus limited to die-cast aluminum parts into which the O-ring grooves can be cast directly. For most of the seals, including those of the engine, a metal backed and molded Viton seal* will be used, as illustrated in Figure 4.9.4. This allows all sealing surfaces to be machined as a flat surface with significant reduction in manufacturing cost, and still insures a good seal. The seal would be preshaped, exactly as a gasket, for rapid assembly on the production line. In design of the engine, the sealed joints have been designed so that only simple "gasket" geometries are required for sealing.

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* Typical of "Gask-O-Seals," manufactured by the Parker Seal Company, Cleveland, Ohio.

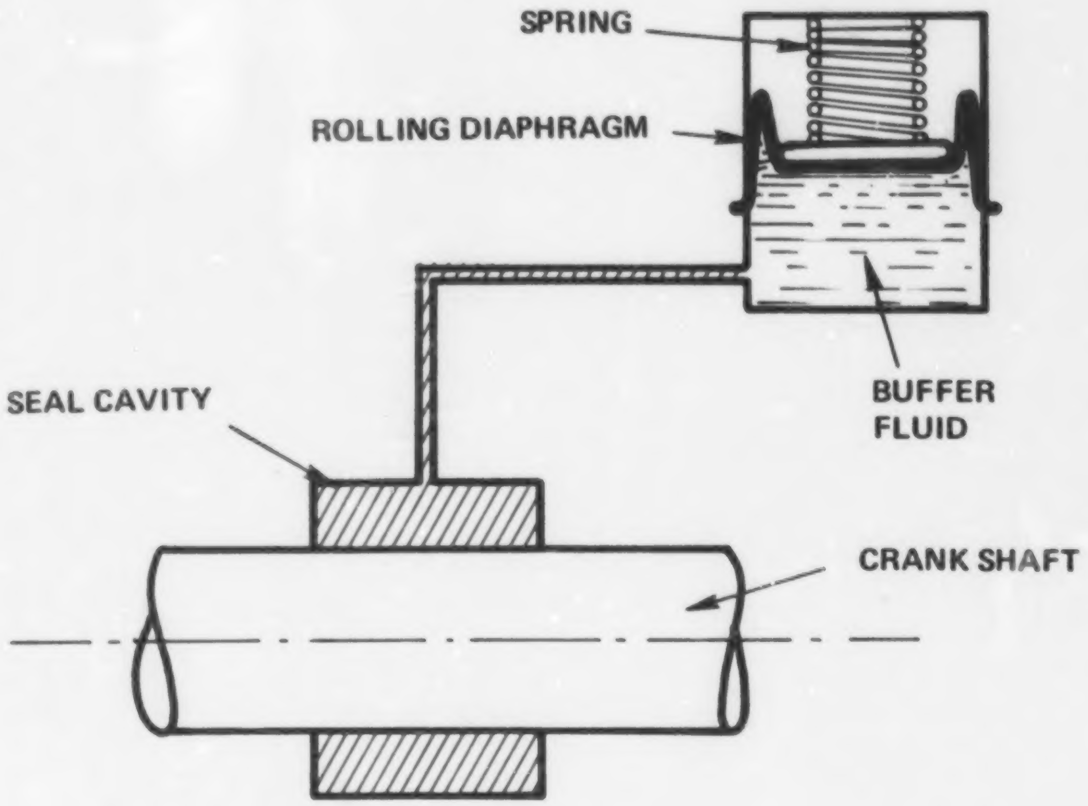
SPRING
ROLLING DIAPHRAGM
SEAL CAVITY
BUFFER FLUID
CRANK SHAFT

Figure 4. 9. 3 Seal Buffer Fluid Accumulator.

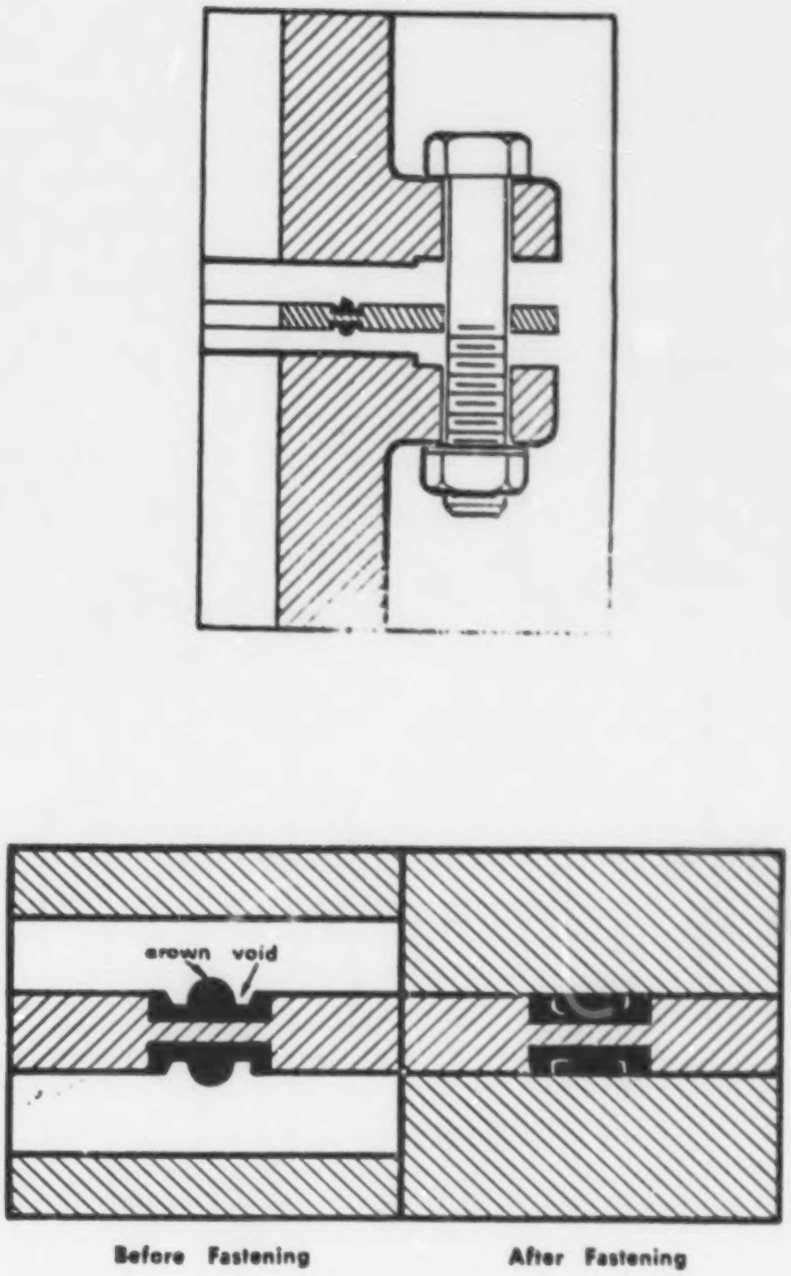
crown
void
Before Fastening
After Fastening

Figure 4.9.4 Static Seal Concept.

4.10 AUTOMATIC TRANSMISSION

A transmission permitting the engine to idle at zero vehicle speed is required if the accessories are to be directly driven by the engine. In addition, a transmission can be used to improve acceleration performance and gradability of a Rankine-cycle propulsion system of a given size. Two transmission approaches have been evaluated in this study. In one, a slipping clutch transmission (either single-speed or two-speed) is used; an overdrive is not required for the approach, since the clutch size is reasonable at the design engine speed and a differential with low ratio can be utilized. In the other, a conventional torque converter is used with approximately 1.88 overdrive (required to reduce the converter size to about 12 inches); a forward-reverse-neutral gear is used after the torque converter. Both approaches are completely automatic with the driver only required to select forward-reverse-neutral-park.

4.10.1 Slipping Clutch

Dana Corporation of Toledo, Ohio, has prepared a design of both a single-speed and two-speed automatic clutch transmission, and the control system for the two-speed transmission has been conceptually prepared. As discussed in Section 5, the two-speed transmission gives much better acceleration performance and gradability than the single-speed and is the preferred approach. The estimated weight of the two-speed transmission is 135 lbs. The design is based on existing technology and is considered state-of-the-art by Dana Corporation. A clutch transmission is completely locked, except at low speeds, giving the equivalent of direct coupling, and should, therefore, be more efficient than a torque-converter transmission.

4.10.2 Torque Converter

The Ford Motor Company has recommended a 12-1/4 inch conventional torque converter for the Rankine-cycle automotove propulsion system with approximately 1.88 overdrive required between the relatively low-speed engine and the converter. The converter must also be integrated with reverse-forward-neutral-park gearing to form a complete transmission. A preliminary assembly drawing of the complete transmission has been prepared at Thermo Electron Corporation using required parts from a manual transmission supplied by the Ford Motor Company.

The torque converter transmission has the important advantage of eliminating high-torque, low speed operation of the engine, thus alleviating potential bearing difficulties in the engine.

4.11 CONTROL AND STARTUP OF SYSTEM

The objective of the control and startup systems selection has been to find the simplest, least expensive and most reliable approach which provides adequate control and startup of the system under all possible types of transients and conditions. The system selected uses mechanical control elements, with diaphragm actuators operated by fluid pressures for modulating control. A complete electrical control system was also evaluated, but was more expensive, more complex and probably less reliable than the mechanical system selected. For a prototype system, an all-electric control system does have the advantage of permitting rapid changes in the control system to improve system control or eliminate control problems encountered in the prototype testing.

4.11.1 Controls for System Operation

In Figure 4.11.1, a complete flow schematic of the system is illustrated, including controls. The schematic does not include the electrical system, startup sequencing controls, ignition system flame sensor or other safety controls.

Probably the most serious control problem in the system is control of the boiler outlet organic pressure and temperature within specified limits regardless of the type of transient encountered by the system. The approach being following is to maintain the burning rate and the feedpump rate to the boiler at values corresponding to the organic vapor flow at any time, that is, to maintain quasi-steady state operation of the boiler. Since practically instantaneous changes from zero to full flow and vice versa can occur, the time delay being the time to depress or release the accelerator pedal, large power

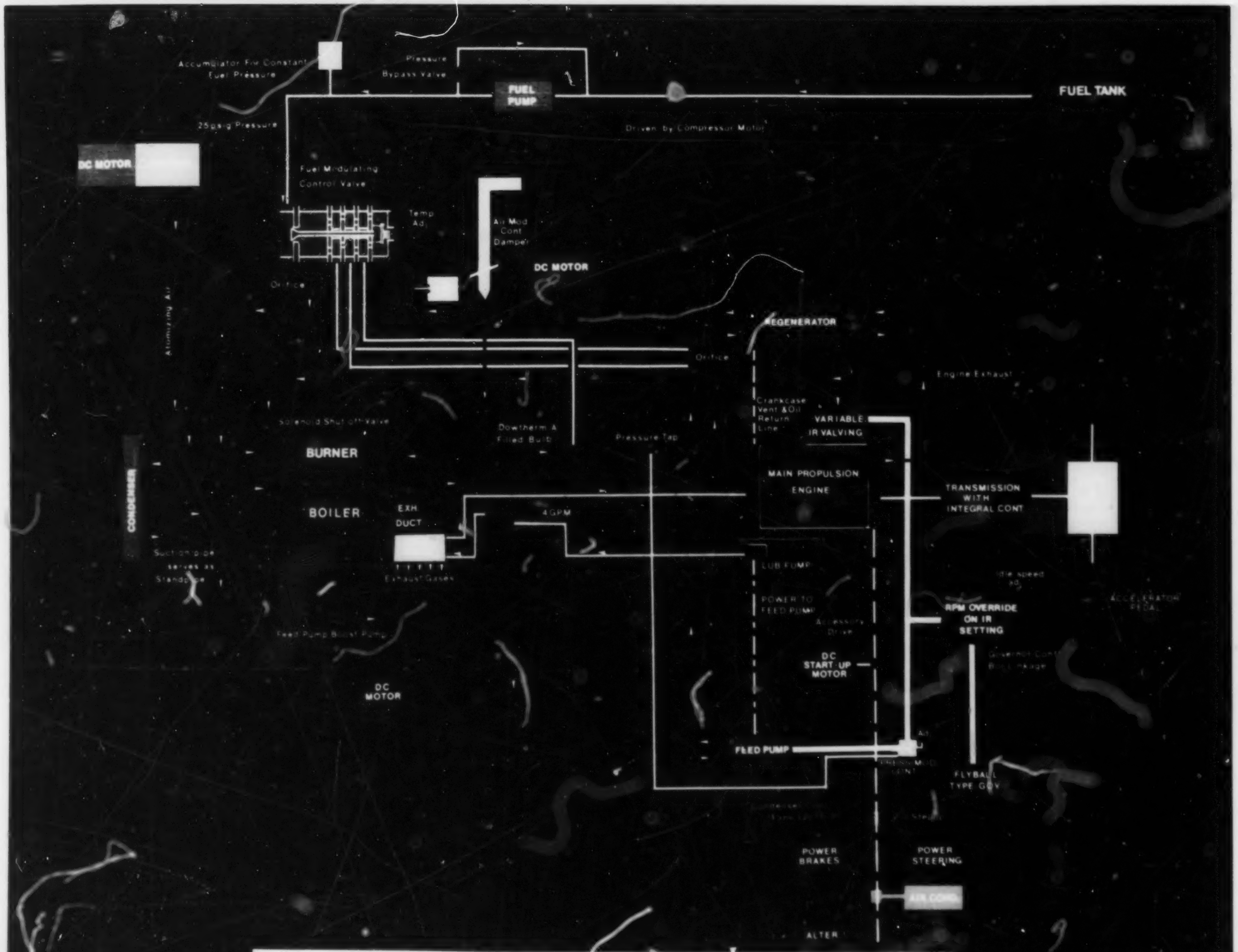
Accumulator For Constant Fuel Pressure
Pressure Bypass Valve
FUEL PUMP
FUEL TANK
25psig Pressure
Driven by Compressor Motor
DC MOTOR
Fuel Modulating Control Valve
Temp Adj
Air Mod Cont Damper
DC MOTOR
Orifice
Atomizing Air
REGENERATOR
Orifice
Engine Exhaust
Crankcase Vent & Oil Return Line
VARIABLE IR VALVING
Solenoid Shut off Valve
Dowtherm A Filled Bulb
Pressure Tap
BURNER
MAIN PROPULSION ENGINE
TRANSMISSION WITH INTEGRAL CONT
CONDENSER
BOILER
EXH DUCT
4GPM
Suction pipe serves as Standpipe
Exhaust Gases
LUB PUMP
POWER TO FEED PUMP
Accessory Drive
Idle speed adj
ACCELERATOR PEDAL
RPM OVERRIDE ON IR SETTING
Feed Pump Boost Pump
DC START UP MOTOR
Governor Cont Bar Linkage
DC MOTOR
FEED PUMP
Adj
PRESS MOD CONT
FLYBALL TYPE GOV
POWER BRAKES
POWER STEERING
AIR COND
ALTER

Figure 4. II. 1 Flow Schematic of System with Controls for System Operation Illustrated Schematically.

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changes in the system must be sensed at the earliest possible time and the burning rate and feedpump rate changed to the new values in a time period of approximately 50 to 100 milliseconds.

The flow rate to the boiler is varied at any engine speed by varying the effective displacement of the feedpump. The control of the feedpump is greatly simplified, however, by the existence of an approximate linear relationship between the mass flow rate through the engine and the intake ratio of the engine, as is evident from Figure 4.11.2. The engine mass flow rate can therefore be expressed as:

$$\dot{m}_{Eng} = C_2 (N_{Eng})(IR)$$

where C_2 = constant
N_{Eng} = engine RPM
IR = engine valve intake ratio

The flow rate of organic from the feedpump can be expressed as:

$$\dot{m}_{FP} = C_1 (N_{FP})(\phi)$$

where C_1 = constant
ϕ = feedpump variable displacement control position.

For equal flow rates and with the feedpump directly driven by the engine with a 1:1 speed ratio,

$$C_2 (N)(IR) = C_1 (N)(\phi)$$

or

$$\phi = \frac{C_2}{C_1} (IR)$$

Preceding page blank

Thus, it may be concluded that the feedpump control position should be directly proportional to the engine intake ratio irrespective of the engine-feedpump speed. The engine intake valve and feedpump variable displacement levers can therefore be connected directly together and operated as a unit. In the schematic, the engine intake valve and feedpump displacement control levers are directly connected to the accelerator pedal through a governor-controlled bar linkage which limits the maximum intake ratio as a function of rpm to prevent exceeding the boiler capacity. The forces required for controlling the hydraulically actuated engine intake and the feedpump displacement should be low enough so that connection can be made directly to the accelerator pedal without power amplification required. Directly connecting both the feedpump and engine to the accelerator pedal insures instantaneous response of the feedpump rate to changes in engine power produced by varying the engine intake valving. In Figure 4.11.3, a schematic representation of the linkage of the accelerator pedal to the system is illustrated.

A diaphragm actuator is also included on the feedpump control to provide a vernier control on the feedpump rate in response to the variable being controlled, the boiler outlet pressure. This actuator uses a spring loaded diaphragm (or bellows) with the boiler outlet pressure applied directly to one side of the diaphragm as illustrated schematically in the flow diagram. Bellow seals will be used in this vernier actuator to eliminate sliding seals, thereby providing a "hermetic" actuator. This control will correct for the relatively small non-linearities between the intake ratio and organic flow rate through the engine, as well as for imbalances which occur between the mass organic flow rates in and out of the boiler (due, for example,

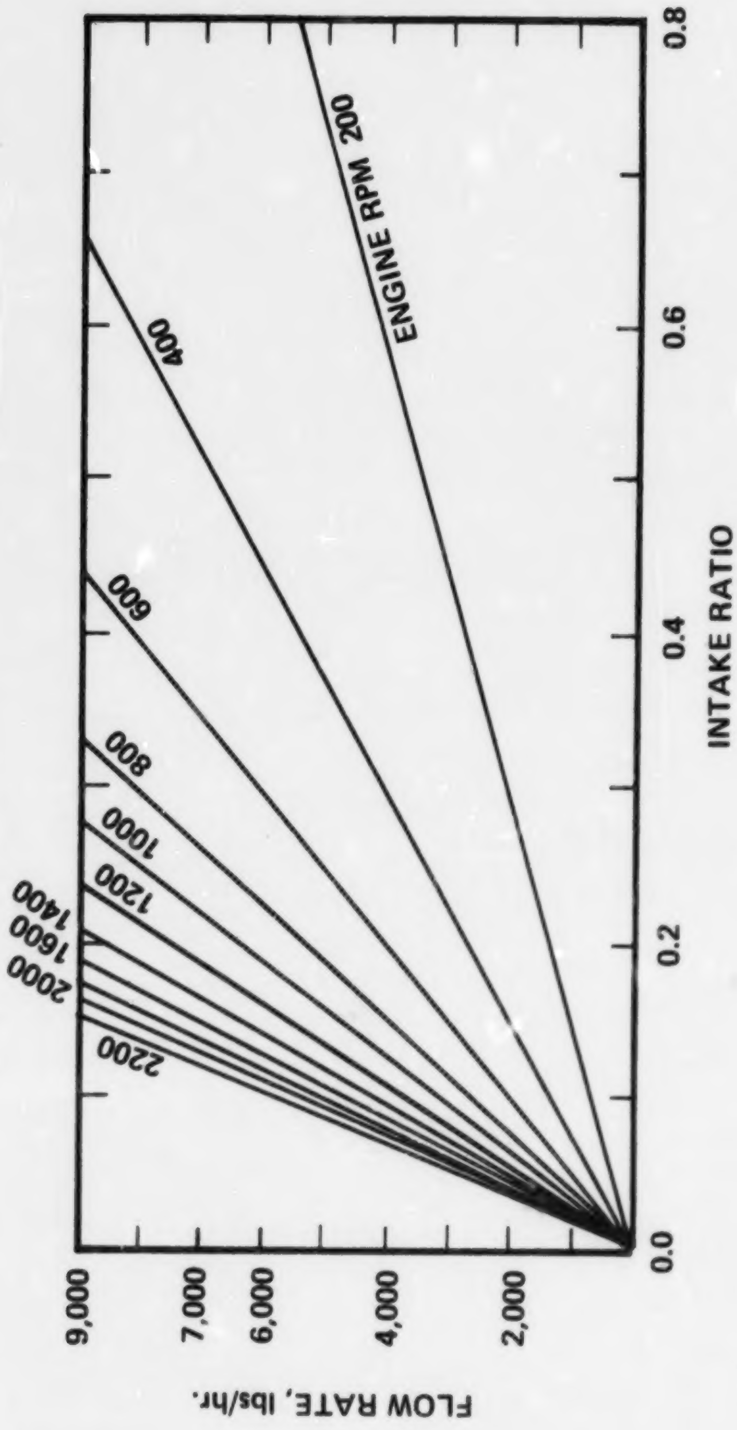


Figure 4.11.2 Mass Flow Rate versus Intake Ratio for Various Engine rpms.

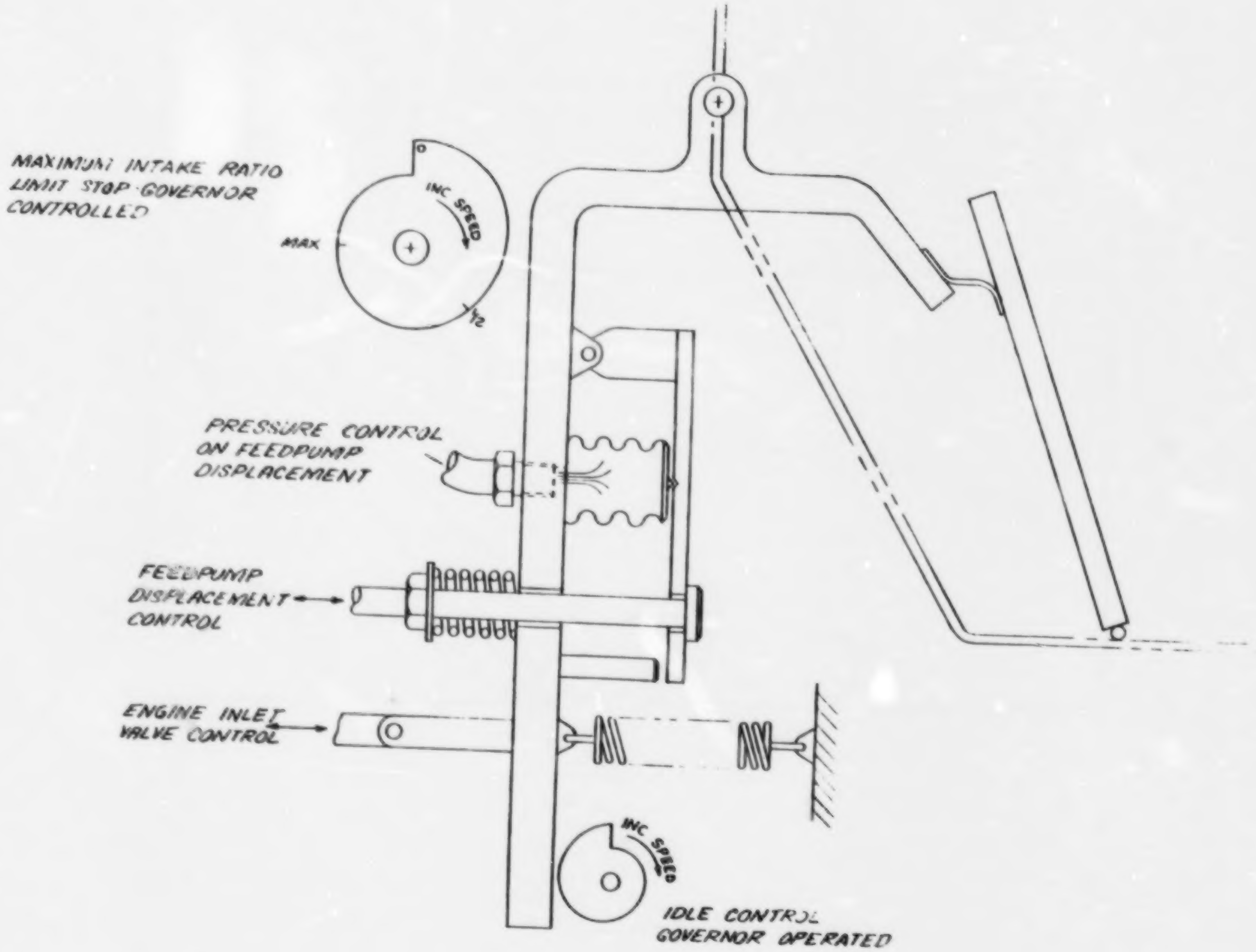

MAXIMUM INTAKE RATIO
LIMIT STOP GOVERNOR
CONTROLLED
MAX
INC SPEED
1/2
PRESSURE CONTROL
ON FEEDPUMP
DISPLACEMENT
FEEDPUMP
DISPLACEMENT
CONTROL
ENGINE INLET
VALVE CONTROL
INC SPEED
IDLE CONTROL
GOVERNOR OPERATED

Figure 4.11.3 Pedal Actuator with Over-Ride and Governor

to momentary loss of pumping due to cavitation) to bring the boiler outlet pressure back to the control point.

The boiler outlet pressure is adjusted by varying the spring force on the vernier controller. On startup, the spring provides full pump displacement until the boiler pressure has increased to 500 psia.

In Figure 4.11.1, the complete burner control is illustrated schematically; a preliminary design of the fuel control valve and actuator in this control system is illustrated in Figure 4.11.4.

The burner control uses an orifice in the organic line to the boiler to detect changes in the organic flow rate to the boiler instantaneously. The orifice ΔP is applied across a diaphragm directly by the thiophene; the force exerted by this ΔP is balanced by the fuel pressure on the discharge side of the fuel valve (fuel inlet pressure is maintained constant); the fuel pressure varies with flow rate by use of an orifice in combination with the fuel nozzle downstream of the valve. As an example of control valve operation, if the organic flow rapidly increases from low flow (low power) to a high flow (full power), a ΔP increase occurs across the orifice. This provides a force unbalance on the valve stem, and the valve opens, increasing the fuel flow rate until the net force on the stem is again balanced. The speed of response of the valve is increased by the fact that the ΔP increase across the orifice is momentarily larger than the steady-state ΔP corresponding to the steady-state flow value. While a detailed analysis of the control dynamics has not been made, extrapolation from similar controllers indicates a speed of response from full closed to full open of the order of 50 - 100 milliseconds. Total travel of the valve is 0.025 inch from closed to full open.

A vernier temperature control similar in concept to the vernier pressure control is also used. A separate spring-loaded diaphragm is used with a pressure proportional to the boiler outlet temperature, provided by a bulb partially filled with Dowtherm A which is immersed in the outlet organic from the boiler. The pressure on the diaphragm is then equal to the vapor pressure of Dowtherm A. In low flow conditions, where speed of response is relatively unimportant, this temperature controller serves as the primary burner control since the orifice ΔP at low flow conditions is low.

As illustrated in Figure 4.11.5, a separate spring-loaded diaphragm actuated by the fuel outlet pressure from the fuel control valve is used to regulate the air flow so that a constant fuel/air ratio is maintained at any burning rate. This actuator regulates the position of a damper in the blower discharge line to the burner.

Both the fuel pump and blower are operated at constant speed by dc motors. Thus, transient response of the burner control is not limited by the need for acceleration or deceleration of these components, but is determined solely by the response of the controller-valve combinations.

Adjustment of the boiler outlet temperature is provided simply by varying the spring force by means of an adjustment screw. On startup, the spring also maintains the fuel valve and air damper in the fully open position for full burning rate until the boiler outlet temperature has reached 550°F.

4.11.2 System Startup

The startup sequencing will be designed for the worst possible condition, thus insuring rapid and positive startup of the system at all times. Difficulties in startup result from two factors:

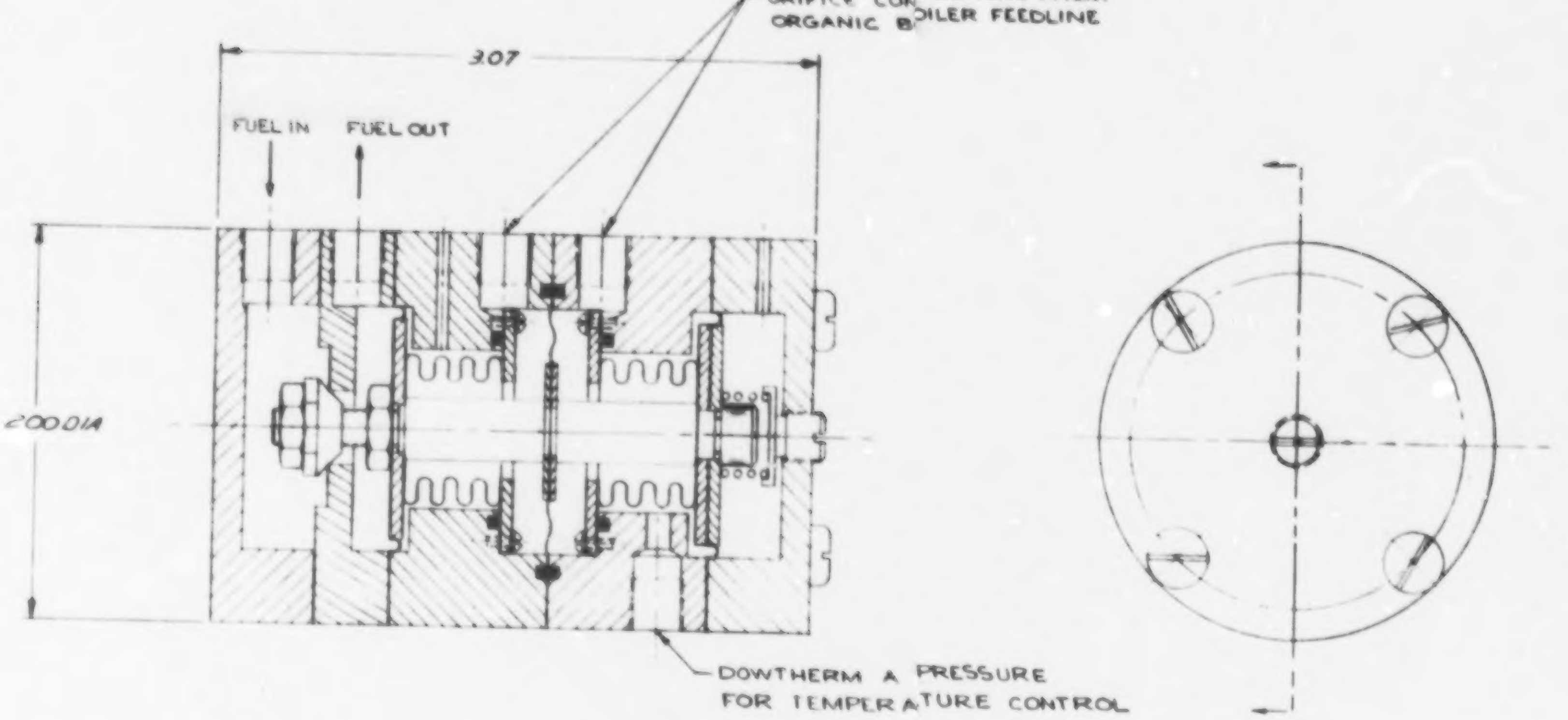

ORIFICE CONNECTIONS FROM
ORGANIC BOILER FEEDLINE
3.07
FUEL IN
FUEL OUT
2.00 DIA
DOWTHERM A PRESSURE
FOR TEMPERATURE CONTROL

Figure 4.11.4 Burner Modulating Fuel Control Valve and Actuator.

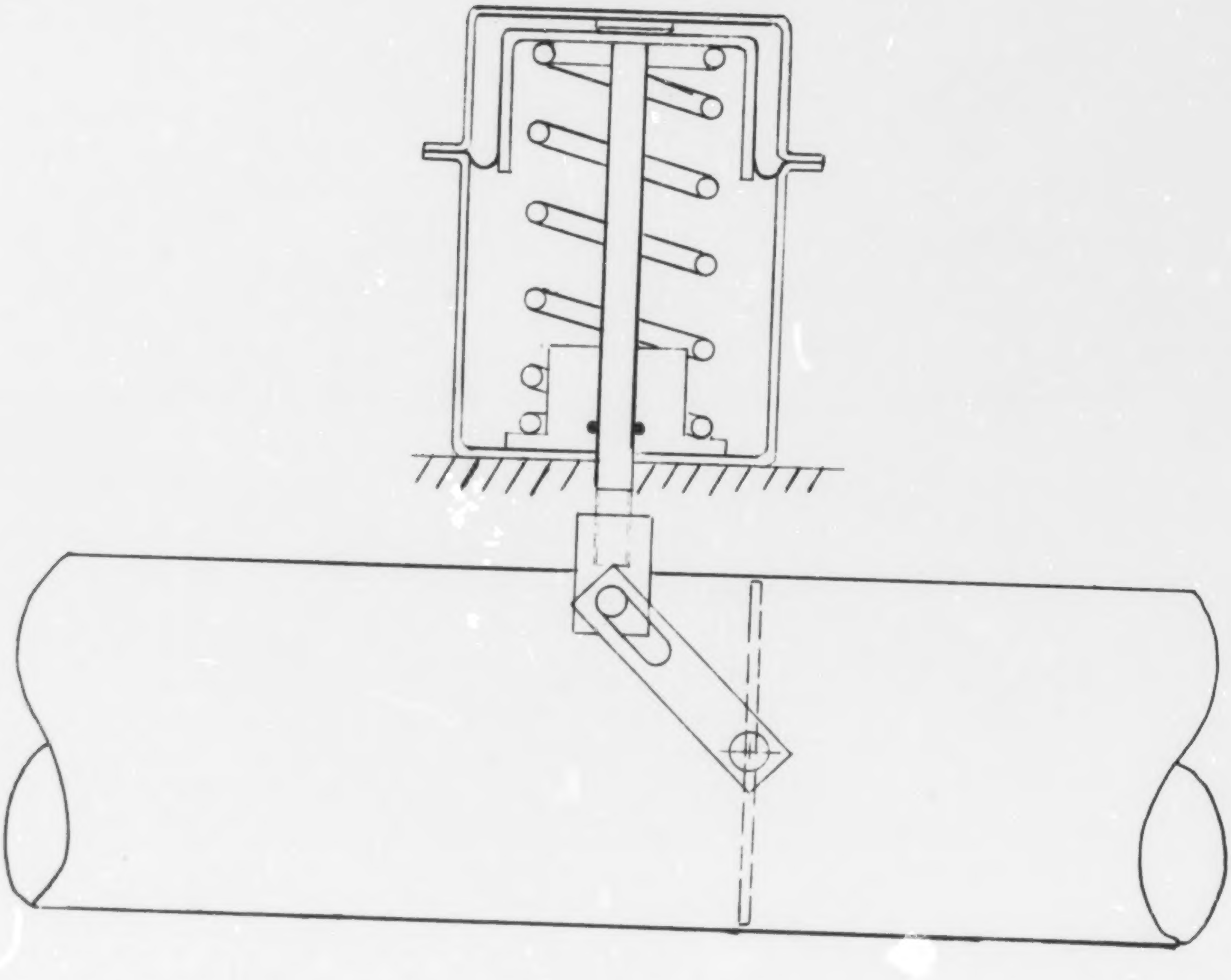


Figure 4.11.5 Air Flow Actuator and Butterfly

(1) In testing at TECO of the piston feedpump for the 3 kwe engine-generator set under development, it has been found that about 1 psi differential is required for opening of the suction valving. On startup in an isothermal sealed system, generation of pressure differentials much less than 1 psi will result in cavitation in the pump, preventing effective pumping. The difficulty is magnified under cold-ambient startup, when the vapor pressure of the thiophene is very low; thiophene has a vapor pressure of 0.14 psia at 0°F.

(2) In a sealed Rankine-cycle system under normal operating conditions, the concentration of thiophene in the miscible lubricant is low because of the high thiophene vapor pressure at the operating lubricant temperature. When the system is shut down and allowed to cool to an isothermal temperature, however, the thiophene from all points in the system will tend to migrate and to dissolve in the lubricant. If the system is shut down for sufficient time, the entire thiophene inventory in the sealed system can migrate to the lubricant in the crankcase so that both lubricant and working fluid are located completely in the engine crankcase. While this migration can theoretically be controlled by shutdown valves which block off parts of the system, failure or leak development in these valves would result in startup failure of the system. The startup procedure proposed is based on the assumption that the entire working fluid-lubricant inventory is in the engine crankcase.

With reference to the flow schematic of Figure 4.11.1, two centrifugal pumps are included in the flow schematic, driven by the same dc motor. One is a feedpump boost pump in the organic line from the condenser to the feedpump section; a standpipe is also included from the condenser outlet to minimize the fluid volume required to provide a liquid head to this pump. This pump will provide a 5 psia pressure rise at the feedpump suction, insuring proper operation of the feedpump suction valving on startup. In normal operation of the system, the pumps are not operating and the feedpump boost pump is designed to pass the full organic flow with negligible pressure drop. The second pump is provided to circulate the liquid in the crankcase through a small finned tube heat exchanger, located in the boiler exhaust, and back to the crankcase. The thiophene will be boiled from the lubricant and vented to the regenerator-condenser side of the flow system. Condensation of the thiophene will occur in the condenser; the condensed liquid will be pushed into the standpipe, providing liquid head to the feedpump boost pump.

A detailed hydrodynamic design of these pumps has been carried out to insure proper operation of the pumps under the very low head conditions which will be encountered in the system startup. In Figure 4.11.6, an assembly cross section of the pumps is illustrated. The pumps are constructed on a common shaft. Since the pressure differential between the two pumps is low (2 psia or less), and since slight leakage between the pumps can be tolerated, the filled fluorocarbon radial and thrust bearing between the pumps is used as the only seal. To eliminate the rotary shaft seal, magnetic coupling across a non-magnetic bore seal is used; the permanent magnet sizes illustrated are sufficient to transmit the relatively low torque required. This type of coupling is currently used on low cost hermetic centrifugal pumps for home hot water heating systems and has been tested at TECO for use on the startup pump for the 3 kwe engine-generator set. The motor is a constant speed (1335), 12 V dc motor generating 0.071 horsepower with 60% efficiency.

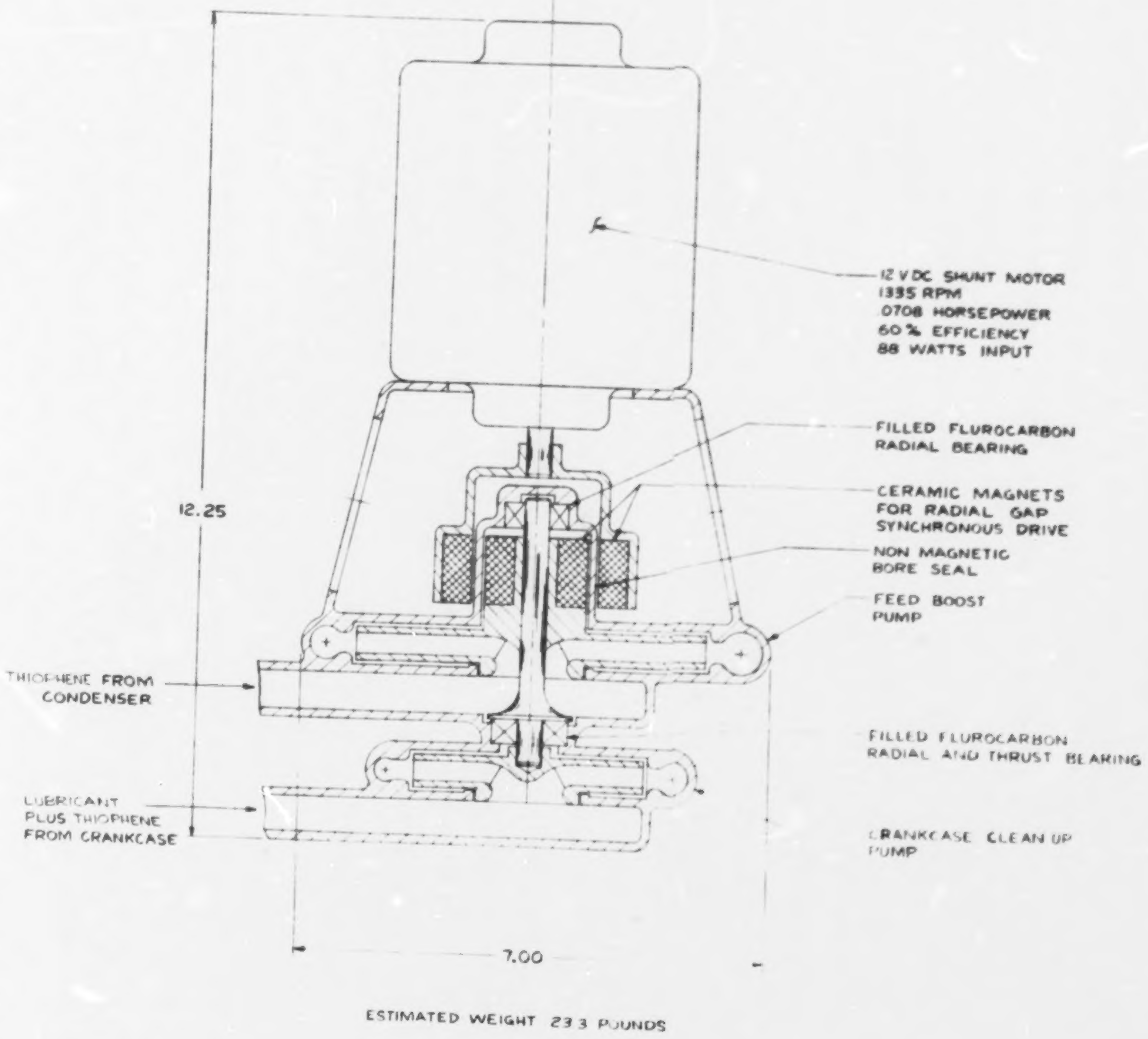

12 V DC SHUNT MOTOR
1335 RPM
.0708 HORSEPOWER
60% EFFICIENCY
88 WATTS INPUT
FILLED FLUROCARBON RADIAL BEARING
CERAMIC MAGNETS FOR RADIAL GAP SYNCHRONOUS DRIVE
NON MAGNETIC BORE SEAL
FEED BOOST PUMP
FILLED FLUROCARBON RADIAL AND THRUST BEARING
CRANKCASE CLEAN UP PUMP
THIOPHENE FROM CONDENSER
LUBRICANT PLUS THIOPHENE FROM CRANKCASE
12.25
7.00
ESTIMATED WEIGHT 23.3 POUNDS

Figure 4.11.6 Startup Pump Assembly.

In Figure 4.11.7, the design operating characteristics of the pumps are illustrated. The feedpump boost pump is designed to provide 7.5 gpm of thiophene flow with 5 psi differential and with a suction head of 7.2 inches of thiophene without cavitation. The crankcase clean-up pump is designed to handle 6 gpm of a fluid composed of 1/3 lubricant and 2/3 thiophene to 100% lubricant with a pressure differential of 2 psia and with a suction head of 5.4 inches without cavitation. Both pumps operate at 1335 rpm with the relatively low speed required because of cavitation.

In Figure 4.11.8, a schematic is presented of a high energy spark generator for ignition, operating off of 12 V dc with a current draw of approximately 1 amp. A breadboard version of this igniter was constructed and operation was satisfactory. The unit uses low-cost standard electronic components and would have overall dimensions of approximately 2 inches by 2 inches by 3 inches. A high energy igniter is very important in minimizing pollutant emission on startup since it is desirable to achieve ignition with the first spray of fuel into the combustion chamber.

In Figure 4.11.9, a schematic is presented of a photoresistor flame sensor to stop fuel flow in case of a flameout. This unit would be integrated with the spark generator and control relays for automatic reignition in case of a flameout. Low cost standard components are used in this unit.

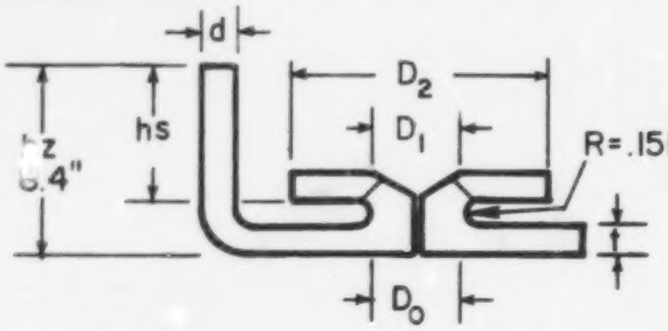
The control and electrical system will be designed for automatic startup of the system initiated by turning of the ignition switch by the operator. The startup sequence is summarized below:

FEED BOOST PUMP

- 7.5 GPM Thiophene
- 5 PSI Pressure Rise Across Pump
- Pass 15 GPM @ Approximately 1 to 2 PSI Pressure Drop Across Starter Pump
- 40 % Efficiency
- 0.0548 Horsepower Input

DETERMINATION OF DESIGN POINT

N = 1.335 RPM
β_1 = 20°
D_1 = 1.09"
D_2 = 5.2"
t = 0.545"
d = 1.54"



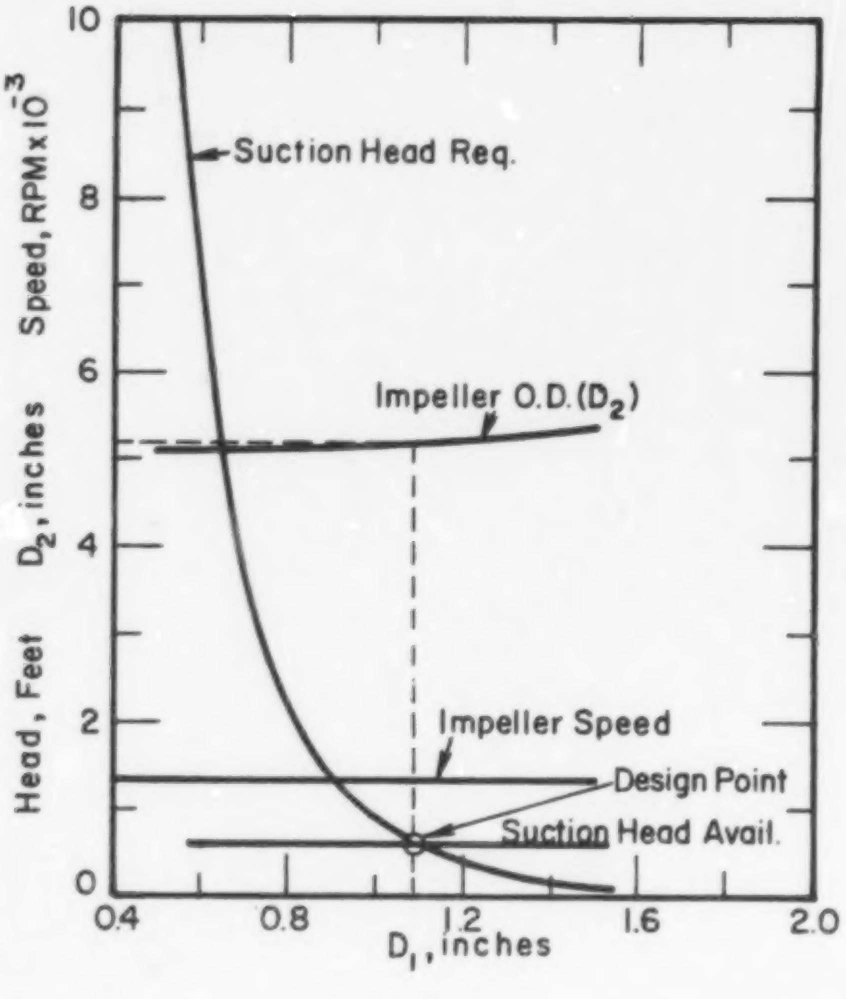


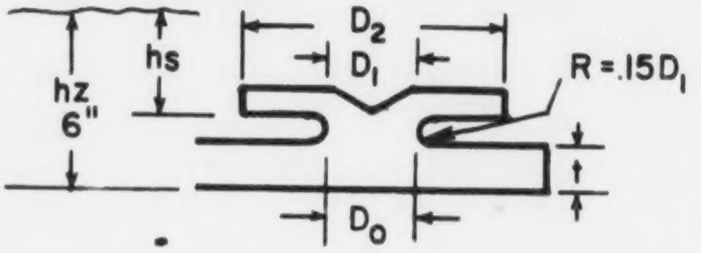
Figure 4.11.7a Characteristics of Feedpump Boost Pump and Crankcase Cleanup Pump Used in Startup of System.

CRANKCASE CLEAN UP PUMP

- 6 GPM (From mixture of $\frac{1}{3}$ Lubricant + $\frac{2}{3}$ Thiophene to all Lubricant)
- 2 PSI Pressure Rise Across Pump
- 40% Efficiency
- 0.016 Horsepower Input

DETERMINATION OF DESIGN POINT

β_1 = 20°
D_1 = 1.05"
D_2 = 3.4"
t = 0.525"
N = 1,335 RPM



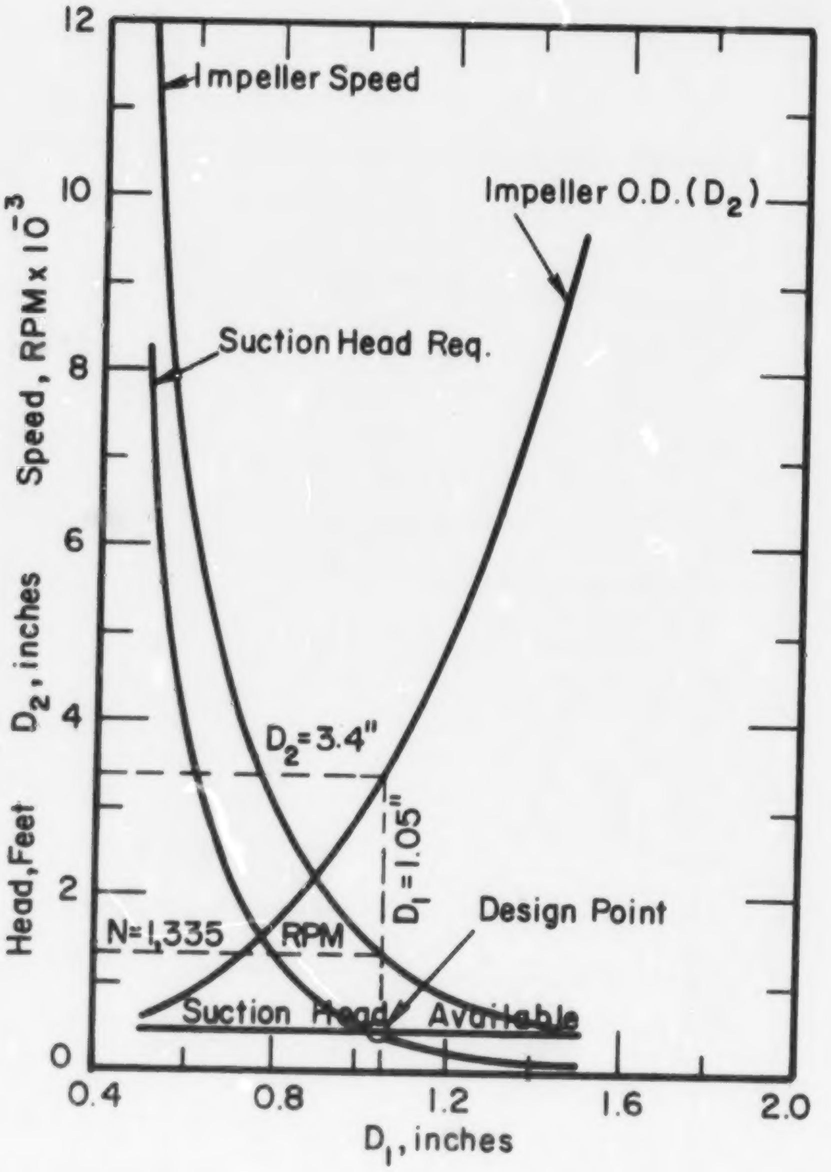


Figure 4.11.7b Characteristics of Feedpump Boost Pump and Crankcase Cleanup Pump Used in Startup of System.

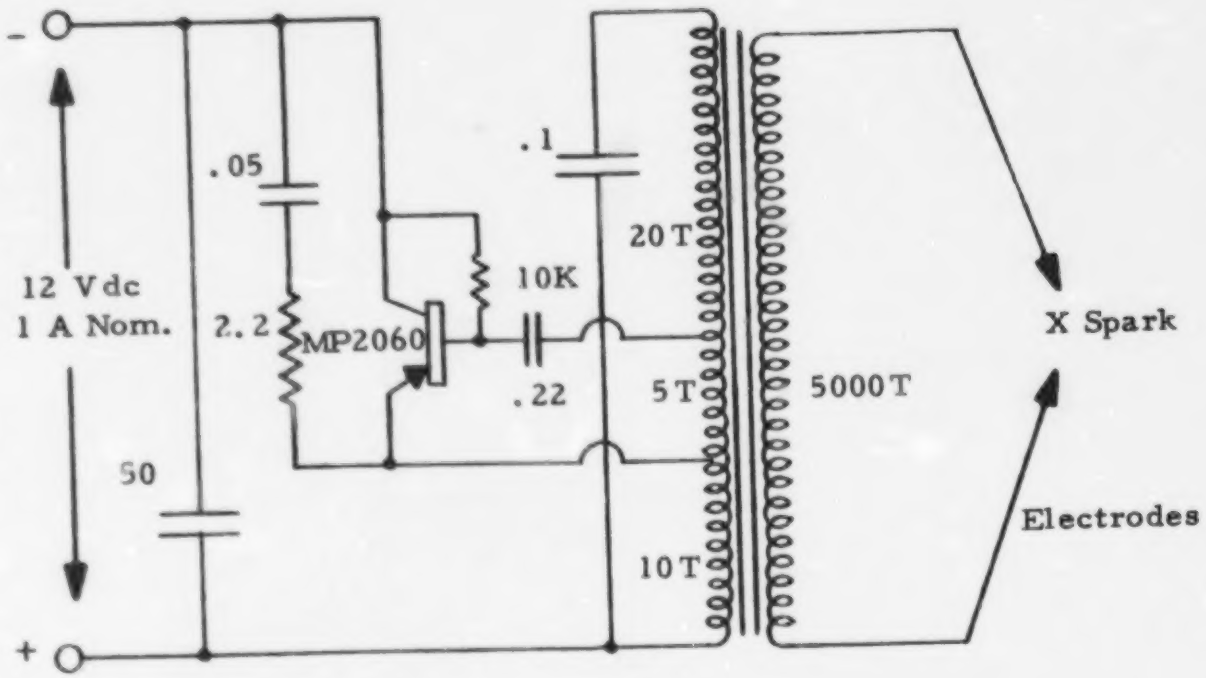

-
+
12 Vdc
1 A Nom.
50
.05
2.2
MP2060
10K
.22
.1
20T
5T
10T
5000T
X Spark
Electrodes

Figure 4.11.8 Igniter - Oscillating Frequency = ~ 30 kc

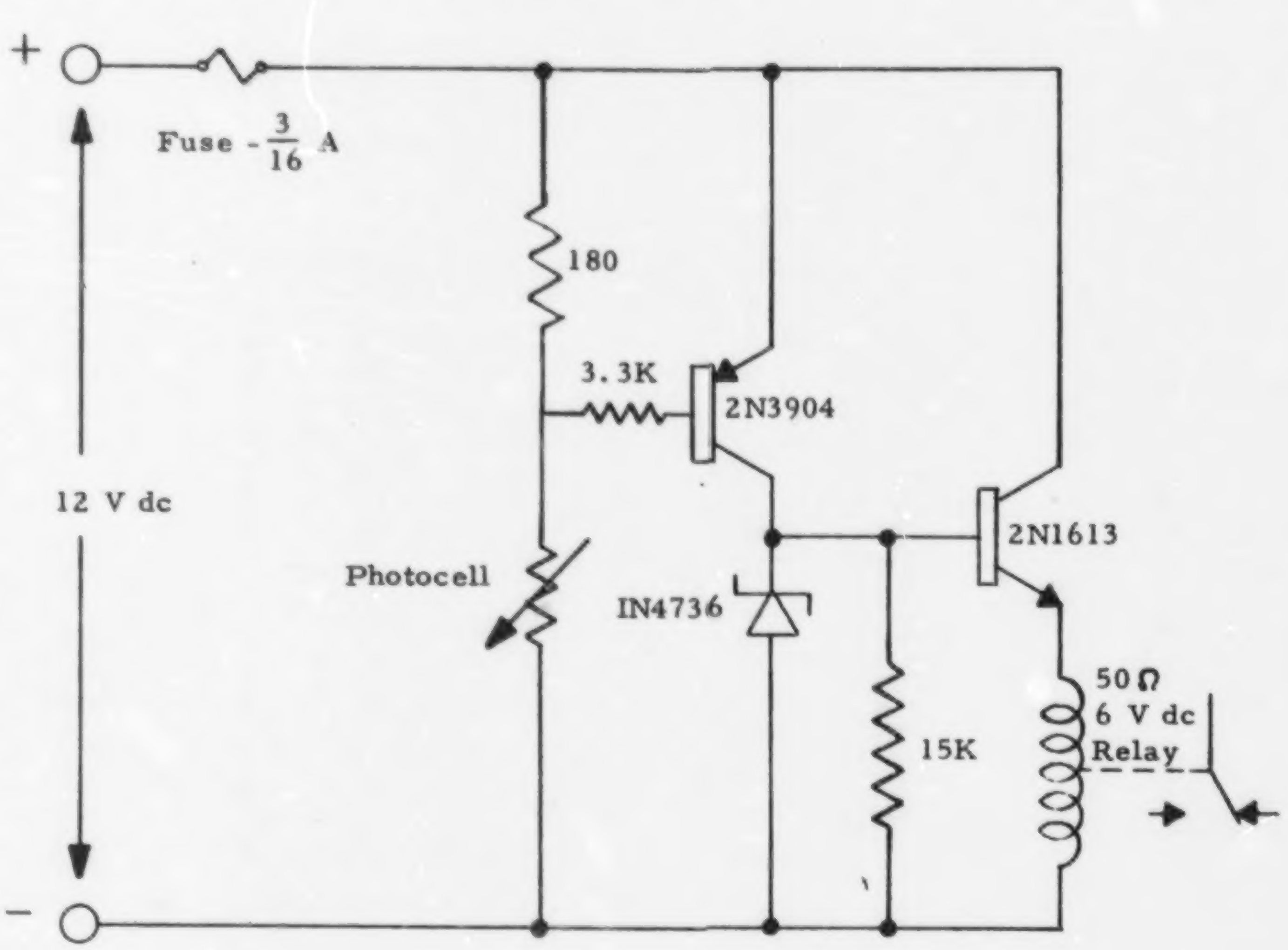

+
-
Fuse - 3/16 A
12 V dc
180
3. 3K
2N3904
Photocell
IN4736
2N1613
15K
50 Ω
6 V dc
Relay

Figure 4. 11.9 Flame Sensor.

a. On turning the ignition switch, the igniter, fuel pump combustion air blower, and startup pumps are turned on.

b. After a short delay (a few seconds) to allow the fuel pump and combustion air blowers to reach operating speed, the fuel shutoff solenoid valve will be opened. If ignition is not achieved in about 5 seconds, the flame sensor will close the solenoid fuel shutoff valve, stopping fuel flow.

c. Approximately 20 seconds after turning the ignition switch on, the engine startup motor will be switched on, rotating the engine, feedpump, hydraulic valve actuating pump and oil lubricating pump. An engine startup speed of about 200 rpm would be used.

d. As boiler pressure builds up, the engine will start, bringing the rpm up to idle speed of 300 rpm.

e. A pressure switch on the boiler will turn off the igniter and startup pumps when boiler pressure reaches approximately 400 psia. This switch will also turn on a green light on the instrument panel, indicating the automobile is ready for operation. On shutdown, turning the ignition switch off will close the solenoid shutoff valve and stop the fuel pump and combustion air blower. The engine will continue to rotate at idle speed until the boiler outlet pressure and temperature decrease as the boiler cools to a level where the accessory and frictional loads stall the engine.

4.11.3 Safety Controls

The following safety and malfunction controls are proposed in the system:

Engine Low Oil Pressure	- Snap-switch operates indicator light for low oil pressure.
Condenser High Pressure	- Snap-switch breaks ignition circuit, shutting system down. - Rupture disc prevents rupture of condenser.
Boiler Water Pressure	
a. High side	- Snap-switch breaks ignition circuit, shutting system down. - Rupture disc prevents tube rupture.
b. Low side	- Rupture disc prevents tube rupture.
Boiler High Organic Pressure	- Relief valve vents liquid from feedpump outlet to condenser. - Snap-switch breaks ignition circuit, shutting system down.
Boiler High Organic Outlet Temperature	- Snap-switch on Dowtherm A for fuel control valve breaks ignition circuit, shutting system down.

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5. SYSTEM DESIGN AND EVALUATION

5.1 INTRODUCTION

A number of alternatives exist with respect to integration of the basic components described in the previous section into a complete system packaged in an automobile. Among the more important choices are:

a. Method of Driving Accessories at Zero Vehicle Speed

In an automotive system, provision must be included for driving the Rankine-cycle accessories as well as the automotive accessories, such as power steering, at zero vehicle speed. In one approach, a relatively simple transmission is used, permitting the main propulsion engine to idle at zero vehicle speed so that all accessories can be driven directly by the main propulsion engine. In the second approach, the main propulsion engine is directly coupled to the drive shaft (through a forward-neutral-reverse gear), with an auxiliary engine running at constant speed used to drive all accessories.

b. Type of Engine Intake Valving

The engine intake valving can be constant intake ratio with a throttle valve for power control or variable intake ratio with power control obtained by adjustment of the engine intake ratio.

c. Packaging of System

Since the Rankine-cycle system consists of several relatively independent components, greater flexibility exists in packaging the system than with an internal combustion engine. For example, the engine could be integrated with the rear axle, with the boiler and the condenser in the normal engine compartment at the front of the car

With respect to driving accessories, use of the main propulsion engine with a simple transmission, permitting the engine to idle at zero vehicle speed, has been selected as the optimum choice. If an accessory engine is used, its size must be sufficient to handle any summation of peak loads by the accessories. Considering all possible accessories, an engine power of approximately 20 hp would be required, even though the average load would be considerably less The engine would therefore be operating at low throttle pressure most of the time, with a poor thermodynamic efficiency. Use of the main propulsion engine permits handling of these peaks so that the additional power required to handle accessories can be based on the average load rather than the peak load. In addition, the power required to drive the accessories will be generated more efficiently by the main propulsion engine. The factors of importance, in addition to efficiency, are cost, packaging, and simplicity. The cost increment represents a tradeoff between the cost of the auxiliary engine-throttle valve control plus smaller accessory components vs. the cost of the transmission plus larger accessory components. This tradeoff is difficult to establish without a detailed design and cost study of all of the components involved, although it would appear qualitatively that the transmission approach cost should be considerably less than that of the auxiliary engine approach. Addition of the 20 hp engine provides a difficult packaging problem if the complete system is to be installed in the engine compartment of a conventional automobile. The required power can be obtained from the main propulsion engine, with negligible increase in size, and the transmission can be packaged exactly as in current IC powered automobiles. The factor of simplicity again represents a tradeoff between the engine-throttle valve control and the transmission, and it is difficult to establish which represents the simpler or more reliable approach.

A computer program has been written for a detailed analysis of the system, using models developed for the various components. Calculations have been used to prepare performance maps for systems with three different types of intake valve operation. In Figures 5. 1. 1 and 5. 1. 2, performance maps are given for a variable intake ratio engine with $(IR)_{max} = 0.8$, selected as the maximum practical intake ratio, and for $(IR)_{max} = 0.29$. The maximum power curve (wide open throttle) on these plots is established by increasing the engine intake ratio at each rpm until the boiler design capacity is reached. In Figure 5. 1. 3, the maximum intake ratio and maximum organic flow rate at which this boiler capacity is reached are illustrated for the two cases. The maximum power curve of Figure 5. 1. 1 with $(IR)_{max} = 0.8$ represents the highest performance for a given engine displacement and boiler capacity. In Figure 5. 1. 4, the maximum torque curves and maximum power curves are presented.

As illustrated in Figures 5. 1. 1 and 5. 1. 2, the maximum engine intake ratio can be decreased from 0. 8 to 0. 29 with a relatively small decrease in performance, since intake ratios higher than 0. 29 can be used only in the range of 300 - 800 rpm, assuming an idle speed of 300 rpm. Use of an intake ratio of 0. 29 results in a significant reduction in the feedpump displacement and condenser load at low engine speeds. From Figure 5. 1. 5, which presents the performance map for a system with throttle valve control and a constant engine intake ratio of 0.137, it is apparent that use of a constant intake ratio with throttle valve control results in a significant decrease in both the efficiency and the performance of the system.

From these performance maps, it is evident that a strong incentive exists for development of an engine with variable intake valving. The choice of $(IR)_{max} = 0.8$ or $(IR)_{max} = 0.29$ is difficult to assess quantitatively since it represents both a cost and a performance tradeoff. In Figure 5.1.6, a comparison is presented of the road load plus grade load for a 1969 Ford Fairlane with the system maximum power output for the cases with $(IR)_{max} = 0.8$ and $(IR)_{max} = 0.29$, respectively. Top speed of the vehicle is seen to be about 100 mph on a level grade. The gradability of the system with $(IR)_{max} = 0.29$ is limited to about a 20% grade assuming a direct drive system. Use of a two-speed transmission can give gradability and performance equivalent to or exceeding that of the system with $(IR)_{max} = 0.8$ when coupled directly.

It is recommended that the development be concentrated on the $(IR)_{max} = 0.29$ valving with two-speed transmission for the following reasons:

(a) Performance and gradability equivalent to or better than that of the $(IR)_{max} = 0.8$ system with one-speed transmission can be obtained by use of a two-speed transmission with $(IR)_{max} = 0.29$.

(b) The condensing load in full-throttle acceleration is easier to meet with a fixed condenser size due to the higher condenser fan rpm at lower vehicle speeds.

(c) Hydrodynamic journal bearing performance in the engine should be more acceptable due to the shorter time for application of full cylinder pressure coupled with the higher engine speed at low vehicle speeds.

(d) The feedpump size is smaller, the required displacement being less by a factor of 2.66.

It should also be noted that engine valving developed for $(IR)_{max} = 0.29$ can be easily extended to $(IR)_{max} = 0.8$ since the primary valving difficulties result from the shorter intake ratios.

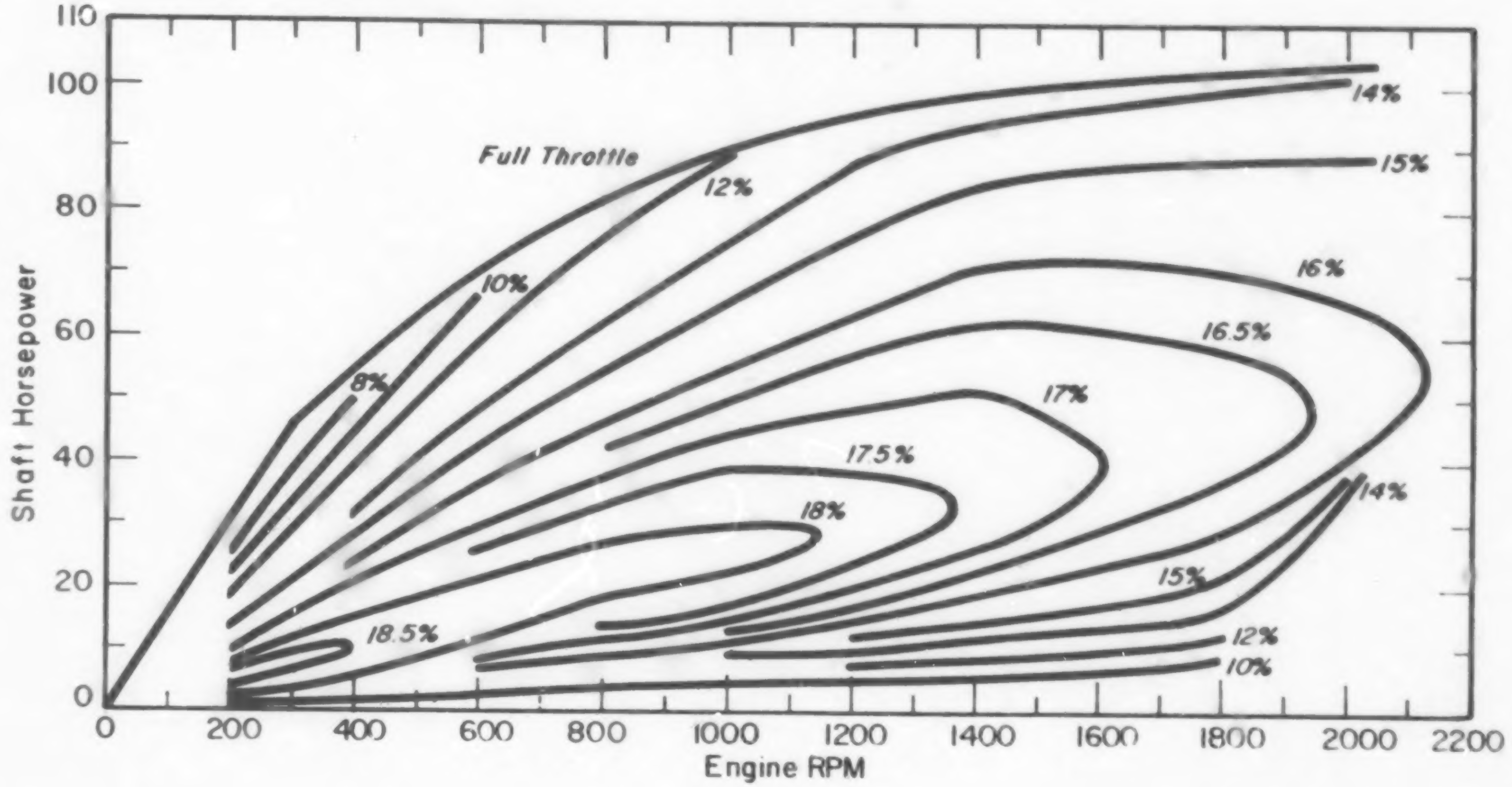

Figure 5.1.1 Performance Map with 184 CID Engine Maximum Intake Ratio of 0.8.

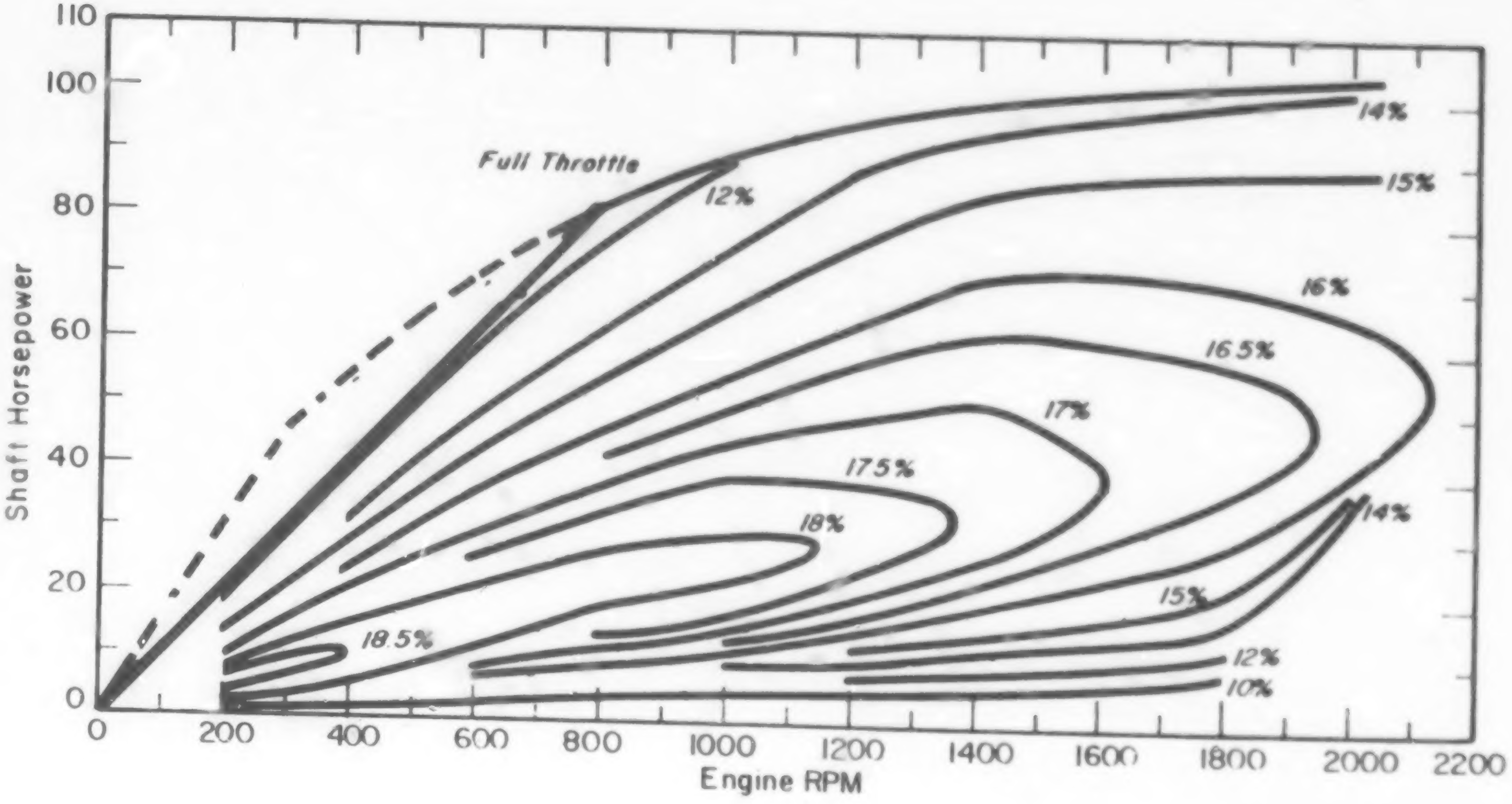

Figure 5.1.2 Performance Map with 184 CID Engine and Maximum Intake Ratio of 0.29.

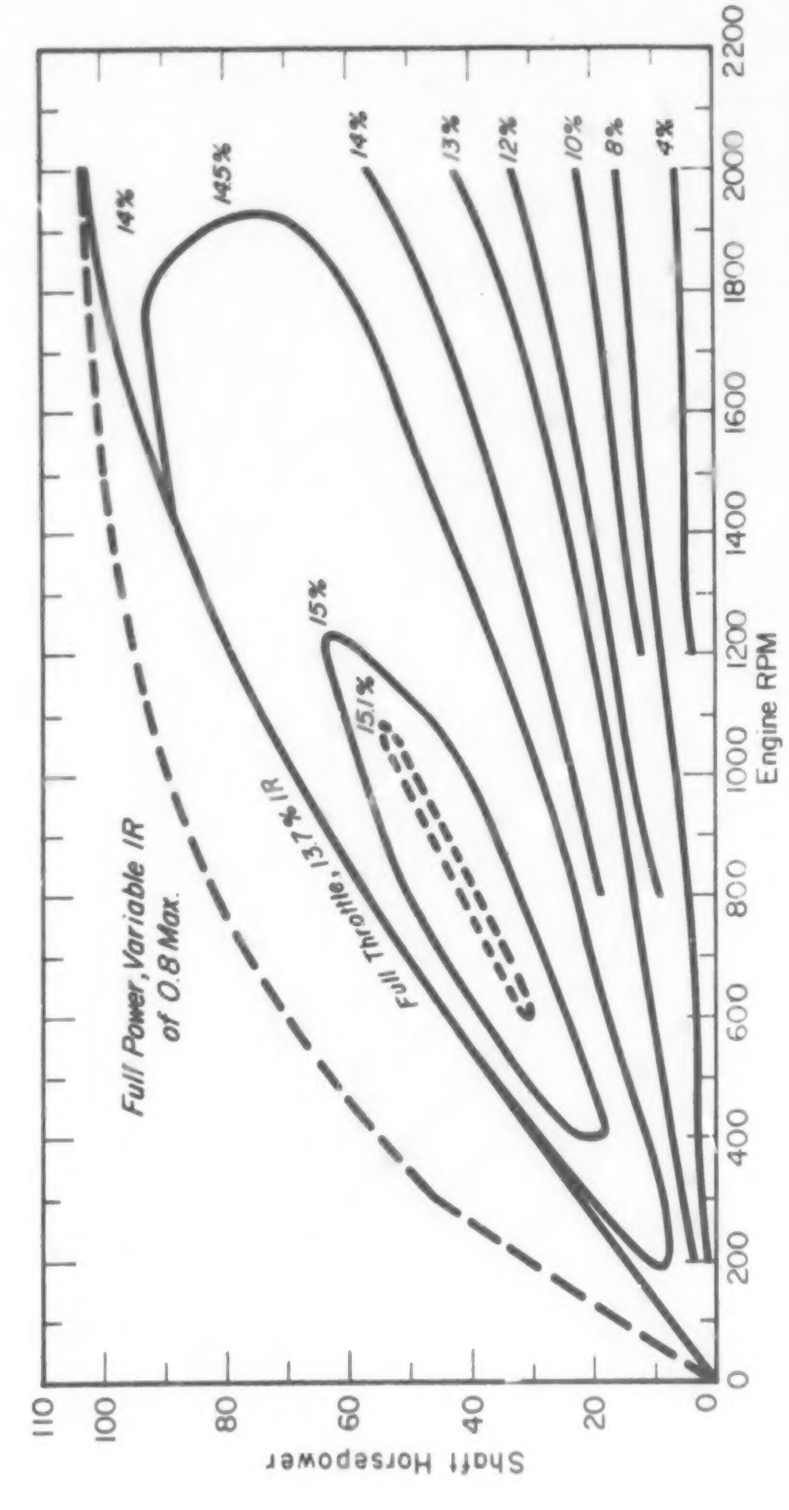

Figure 5.1.3. Performance Map with 184 CID Engine and Constant Intake Ratio of 0.137 (Throttle Valve Control)

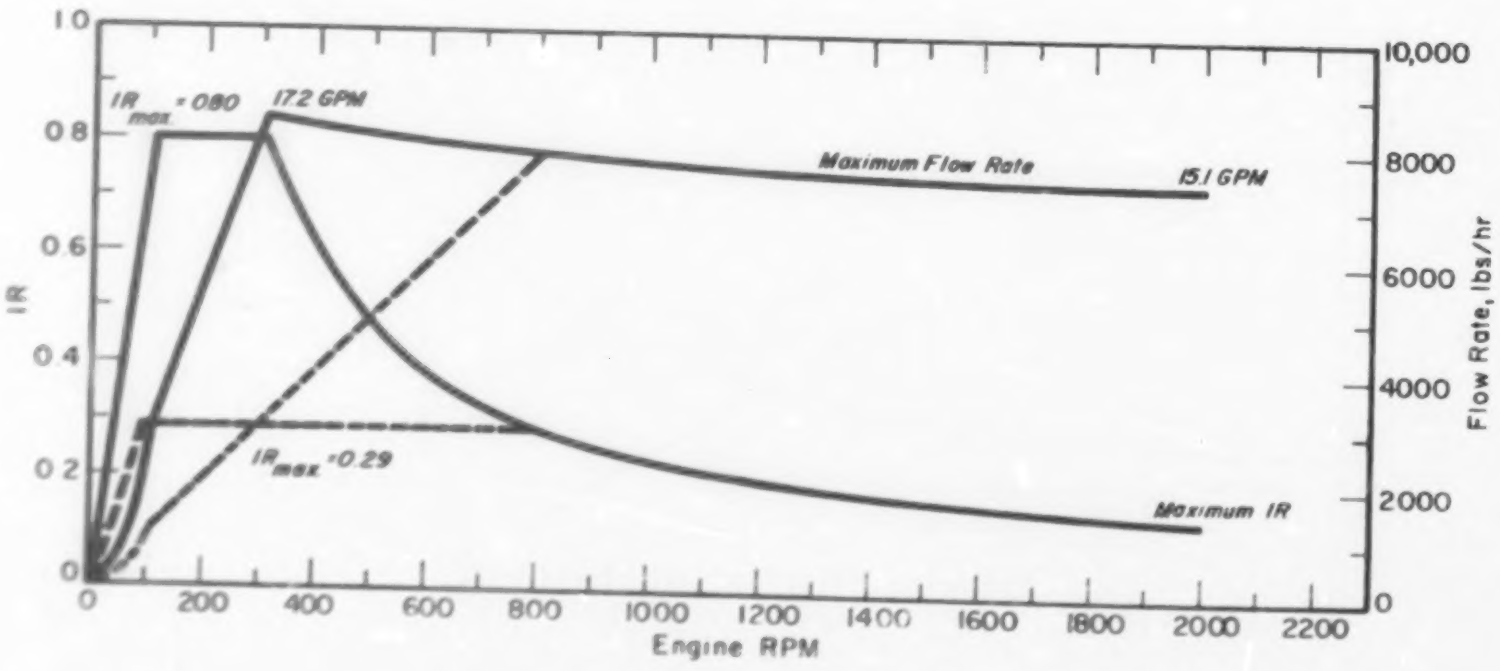

Figure 5.1.4 Maximum Intake Ratic and Maximum Organic Flow Rate as Functions of Engine Speed.

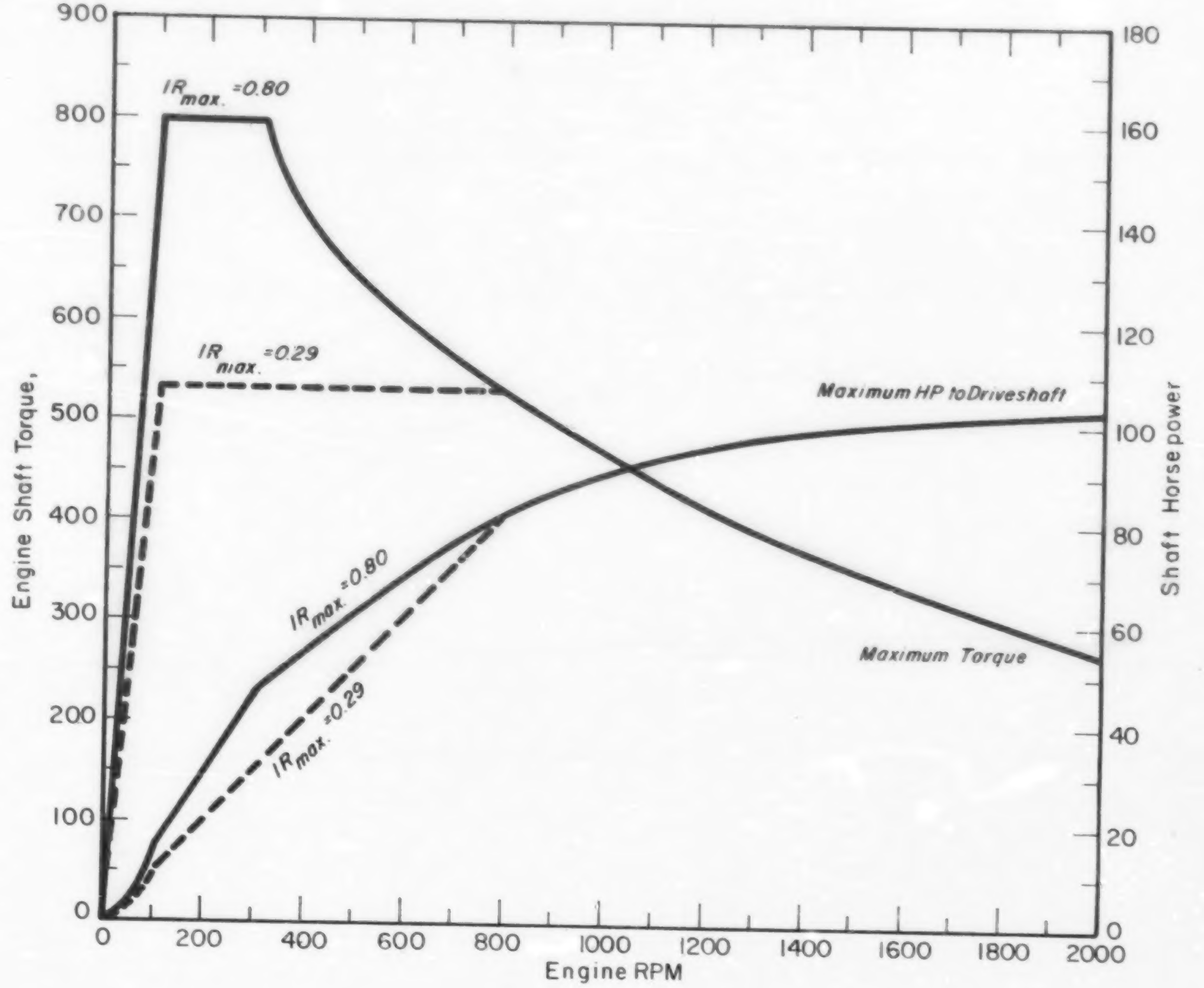

Figure 5.1.5 Comparison of Maximum Horsepower and Maximum Torque for $IR_{max} = 0\ 8$ and $IR_{max} = 0.29$.

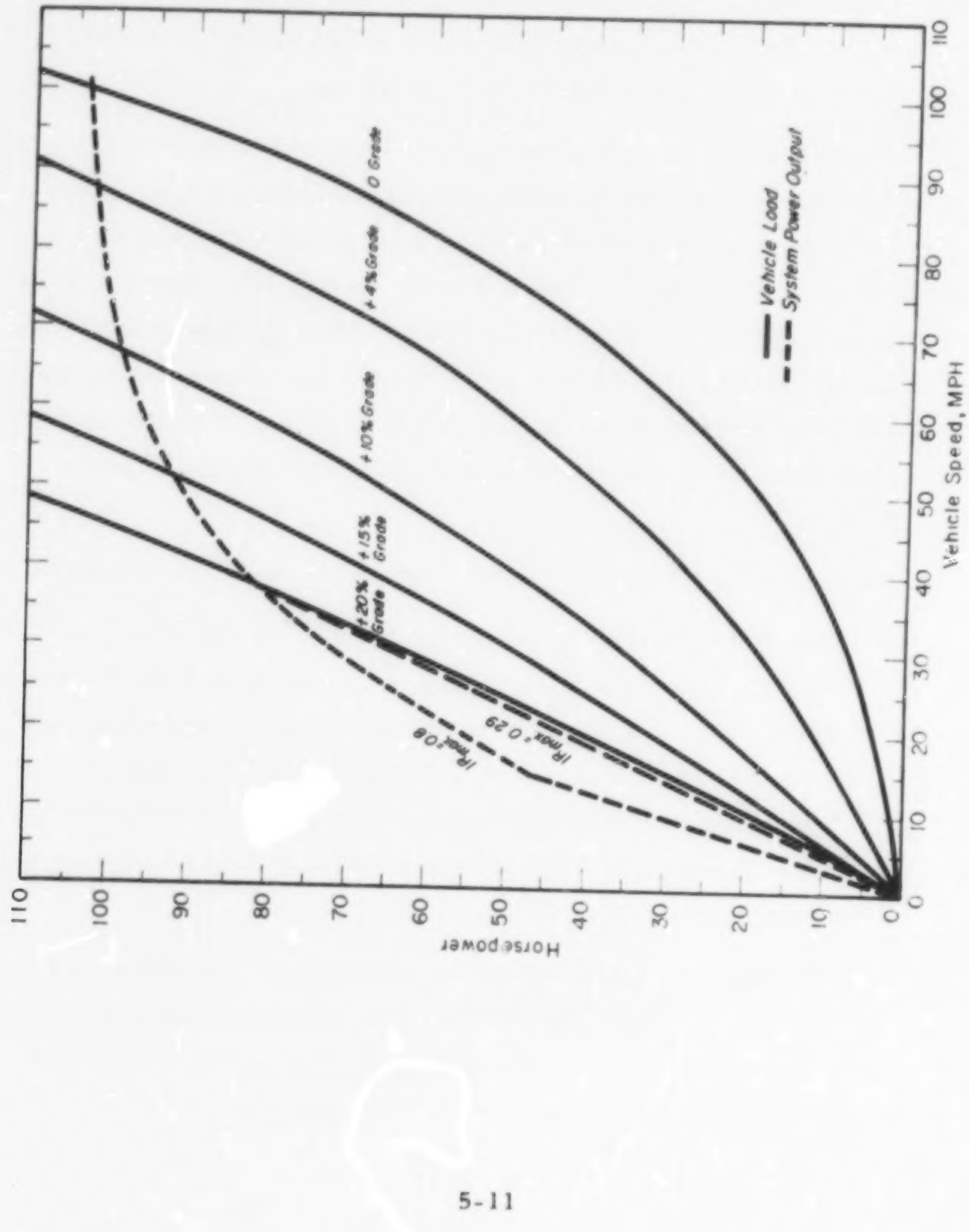

Figure 5.1.6 Comparison of Road Load Plus Grade Load for 1969 Ford Fairlane with System Maximum Power Output.

5.2 PERFORMANCE IN REFERENCE AUTOMOBILE

A 1969 4-door Ford Fairlane has been selected as the reference automobile for this study. Performance characteristics of the Rankine-cycle system are presented in this section for a torque converter transmission and single-speed clutch transmission with $IR_{max} = 0.8$ and for single and two-speed clutch transmissions with $IR_{max} = 0.29$.

Figures 5.2.1 through 5.2.4 and Tables 5.2.1 through 5.2.5 present the performance and fuel economy calculations prepared by the Ford Motor Company and the Dana Corporation, using existing computer programs, for the Rankine-cycle system with different types of transmissions and for the Ford production 302 - 2V engine with three speed transmission. The basic input to these calculations was the performance maps presented in Figures 5.1.1 and 5.1.2. Fuel economy calculations have been completed only for the Rankine-cycle system with $IR_{max} = 0.8$.

The most important conclusions from these calculations are:

1. The Rankine-cycle system with 184 CID engine should be capable of providing 0 - 60 mph acceleration times of less than 15.0 seconds, taken as the criterion for acceptable performance. The acceleration performance is fairly dependent on the type of transmission used. A maximum level grade vehicle speed of 95 - 100 mph should be attainable, irrespective of type of transmission used.
2. The Rankine-cycle system with two speed clutch transmission and $IR_{max} = 0.29$ provides a close approximation to the tractive effort delivered by the 302 - 2V internal combustion engine with three speed automatic transmission.

3. The Rankine-cycle system with two-speed clutch transmission and $IR_{max} = 0.29$ provides a gradability of 49% with 54 ft-lbs of torque subtracted for driving accessories.

4. To obtain performance with the Rankine-cycle system, with $IR_{max} = 0.8$ and single-speed clutch transmission, equivalent to that of the 302 - 2V internal combustion powered system would require an increase in the size of the Rankine-cycle system of about 20% (engine displacement = 220 CID, with appropriate increases in boiler, condenser, feedpump, and other accessory sizes).

5. The customer average fuel economy for the Rankine-cycle system with $(IR)_{max} = 0.8$ and single-speed clutch transmission is 24% less than the 302 - 2V internal combustion system (15.7 mpg vs 12.7 mpg).

6. The suburban fuel economy for the Rankine-cycle system with $(IR)_{max} = 0.8$ and single-speed clutch transmission is 14% less than the 302 - 2V internal combustion system (18.0 mpg vs 15.8 mpg).

7. The city fuel economy for the Rankine-cycle system with $(IR)_{max} = 0.8$ and single-speed clutch transmission is 39% less than the 302 - 2V internal combustion engine. It is expected that this comparison will be improved by use of the Rankine-cycle system with $IR_{max} = 0.29$ and two-speed clutch transmission, since this eliminates the region of lowest system efficiency in the low speed range and moves the overall system operation into a more efficient region for the Rankine-cycle system at low vehicle velocities (see performance maps, Figures 5.1.1 and 5.1.2).

(8) The steady speed fuel economy for the Rankine-cycle system with $IR_{max} = 0.8$ and single-speed clutch transmission is higher than that of the 302 - 2V internal combustion system up to 50 mph and is lower above this speed.

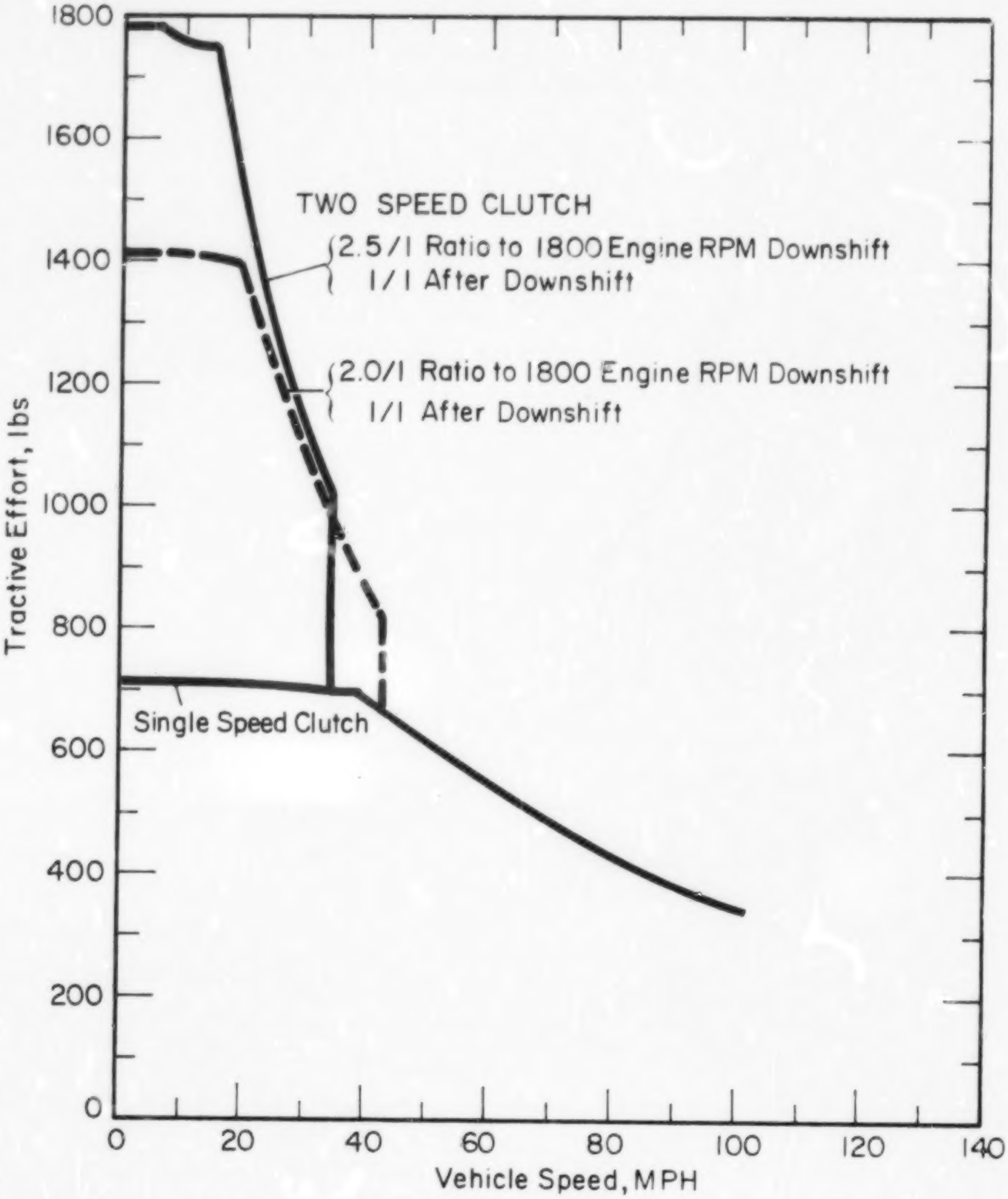

Figure 5. 2. 1 Comparison of Tractive Effort for Single and Two Speed Clutch Transmission.

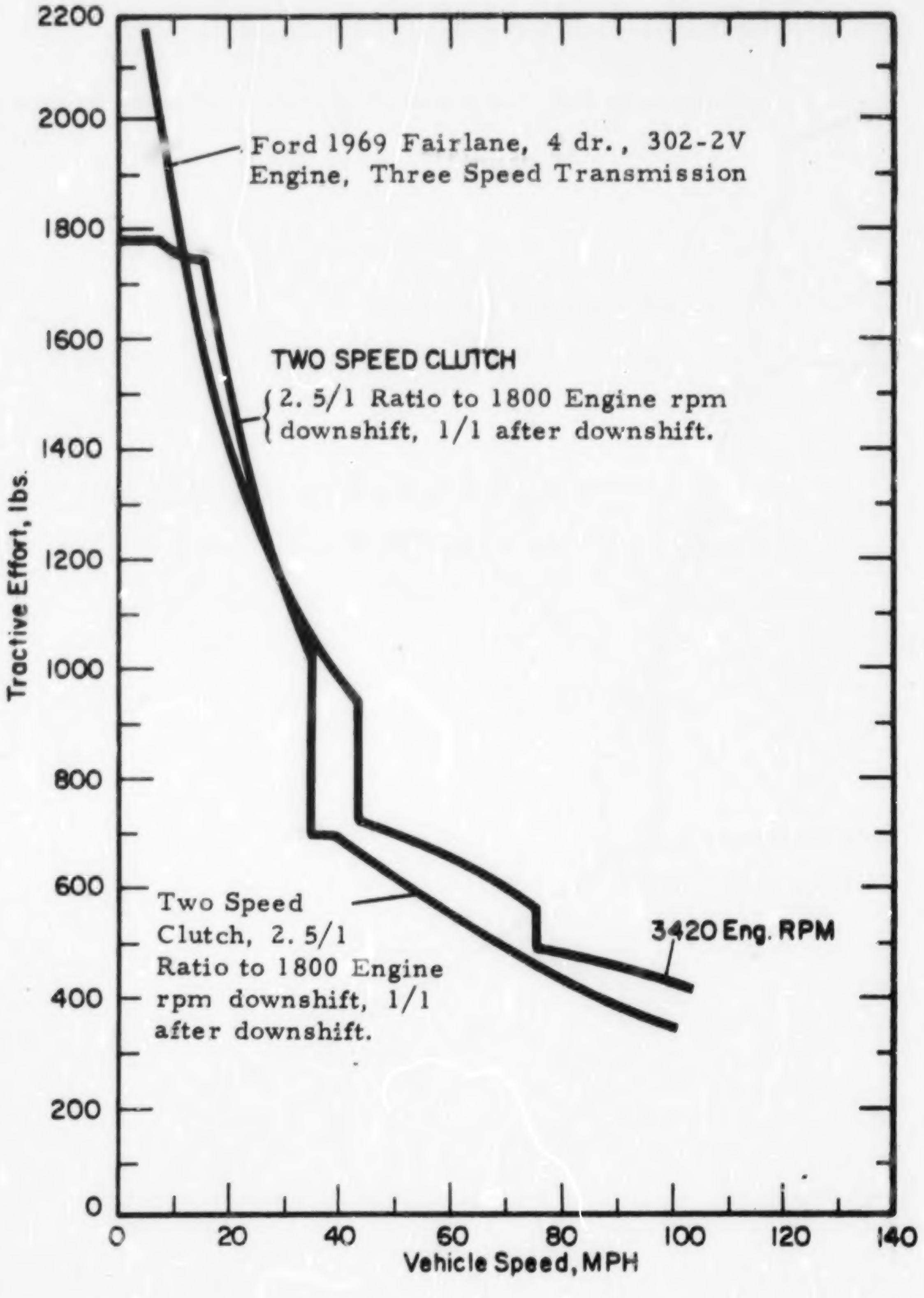

Figure 5.2.2 Comparisons of Tractive Effort for 1969 Ford Fairlane Powered by 302-2V Engine and by Rankine-Cycle System with 184 CID Engine, $IR_{max} = 0.29$, with Dana Two Speed Clutch Transmission.

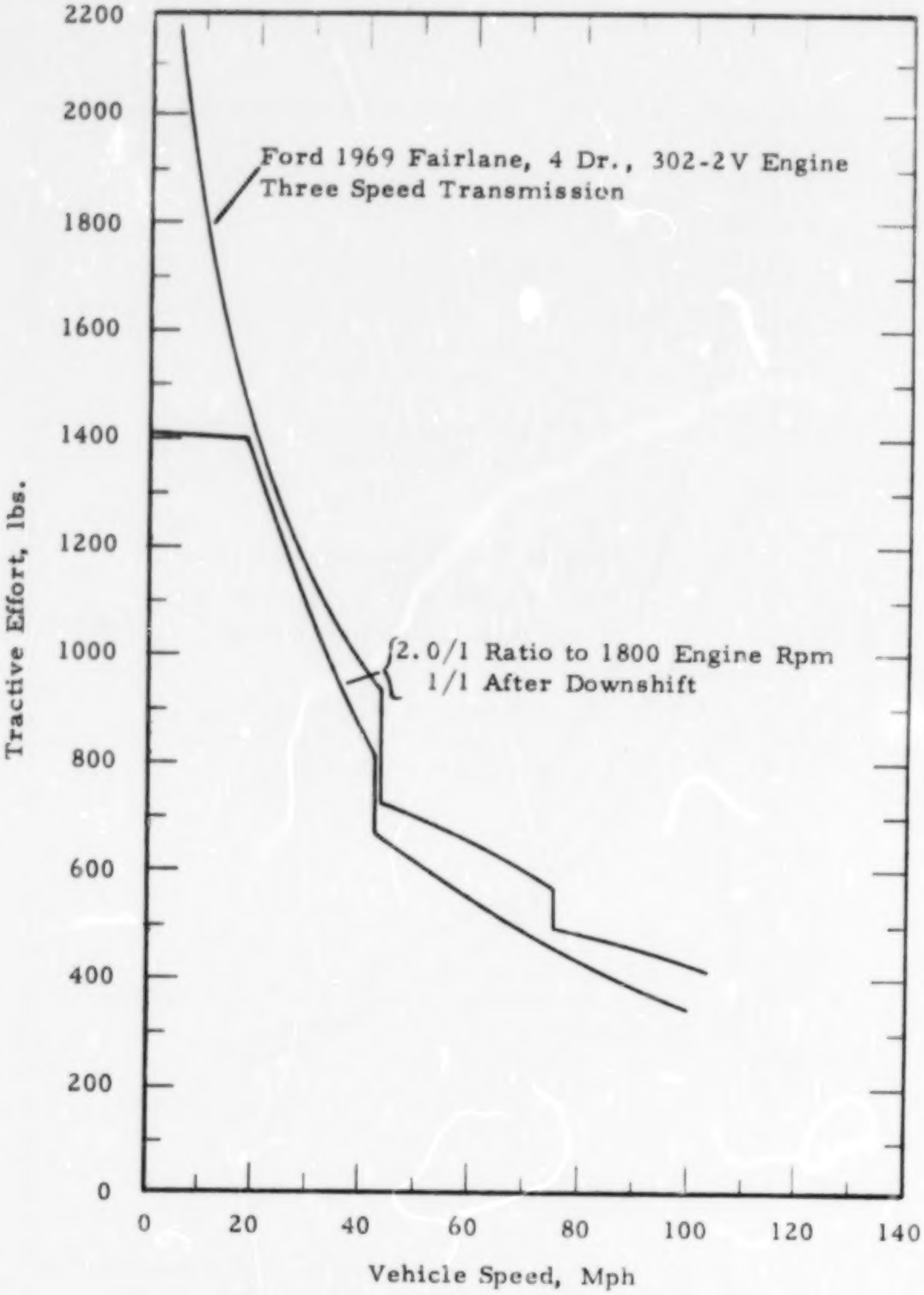

Figure 5.2.3 Comparison of Tractive Effort for 1969 Ford Fairlane powered by 302-2V Engine and by Rankine-cycle system with 184 CID Engine, $IR_{max} = 0.29$, with Dana two-speed clutch transmission.

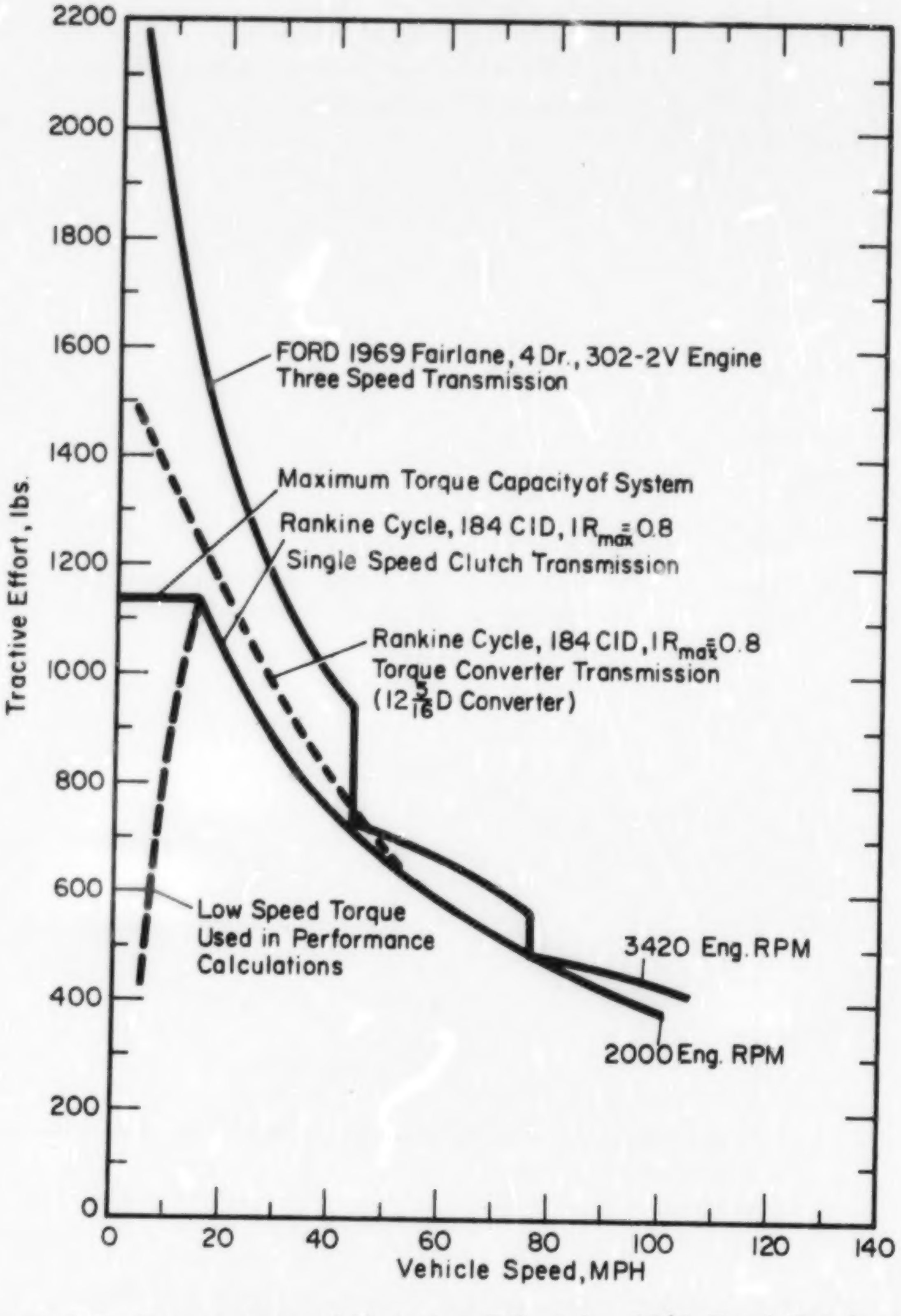

Figure 5.2.4 Comparison of Tractive Effort for 1969 Ford Fairlane Powered by 302-2V Engine and by Rankine-Cycle System with 184 CID Engine, $1R_{max} = 0.8$, with Single Speed Clutch Transmission or with Torque Converter Transmission.

TABLE 5.2.1

SUMMARY OF RANKINE-CYCLE ENGINE PERFORMANCE AND ECONOMY PROJECTIONS FROM COMPUTER PROGRAMS PB1111 AND PB1213

$(IR)_{max} = 0.8$

VEHICLE: 1969 Fairlane 4-Door Sedan	PERFORMANCE PB1111				ECONOMY PB1213			
	0-4 Sec. Ft.	0-10 Sec. Ft.	0-60 mph Sec.	Passing at 50 mph Sec.	City mpg	Suburban mpg	Customer Average	70 mph mpg
ENGINE AND DRIVE								
302-2V Production Engine Automatic Transmission	90	469	11.9	9.7	13.3	18.0	15.7	16.2
250-1V Production Engine Automatic Transmission	78	406	15.6	11.6	12.9	19.6	16.3	17.59
182-CID TECO Engine, Single Speed Clutch Drive	70	408	14.2	10.3	9.6	15.8	12.7	13.4
220-CID TECO Engine Single Speed Clutch Drive	86	480	11.4	9.2	-	-	-	-
182-CID TECO Engine Speed-up 1.88, Converter	74	412	14.4	11.0	10.1	15.0	12.5	13.0

TABLE 5.2.2

VEHICLE PERFORMANCE PROJECTIONS
THERMO ELECTRON CORPORATION
RANKINE CYCLE ENGINE

VEHICLE: 1969 Fairlane 4-Door Sedan, Wheelbase - 116 in.
TIRES: 7.75 x 14, Rolling Radius - 1.08 ft., Rev/Mile - 778

$(IR)_{max} = 0.8$

	Transmission Gear Ratio	Axle Ratio	N/V	Car Weight lbs.	Performance - Computer Program PB1111				
					0 - 4 Sec. ft.	0 - 10 Sec. ft.	0 - 60 mph sec.	0 - 1/4 Mile sec.	Passing at 50 mph sec.
Ford Production Engines									
302-2V 3-Speed Automatic 11-1/4 Dia. Converter	2.46, 1.46, 1.00	2.79	36.2	3539	89.9	469.2	11.9	18.8	9.72
250-1V 3-Speed Automatic 11-1/4 Dia. Converter	2.46, 1.46, 1.00	2.79	36.2	3522	78.2	405.7	15.6	20.6	11.59
Rankine Cycle Engines									
182-CID Clutch Drive, Single Speed	None	1.64	21.3	3539	70.0	407.9	14.2	20.3	10.30
200-CID Clutch Drive, Single Speed	None	1.64	21.3	3539	77.2	442.0	12.7	19.5	9.74
220-CID Clutch Drive, Single Speed	None	1.64	21.3	3539	86.0	480.4	11.4	18.7	9.22
240-CID Clutch Drive, Single Speed	None	1.64	21.3	3539	94.4	516.7	10.3	18.0	8.79
182-CID 1.88 Ratio Speed-Up Gear 12-5/16 Dia. Converter	None	3.06	21.3	3539	74.2	412.2	14.4	20.1	10.97
182-CID 2.77 Ratio Speed-Up Gear 12-5/16 Dia. Converter	None	4.50	21.3	3539	79.8	414.6	14.3	20.1	10.85

TABLE 5.2.3

VEHICLE ECONOMY PROJECTIONS
THERMO ELECTRON CORPORATION
RANKINE CYCLE ENGINE

VEHICLE: 1969 Fairlane 4-Door Sedan, Wheelbase - 116 in.
TIRES: 7.75 x 14, Rolling Radius - 1.08 ft., Rev/Mile - 778

$(IR)_{max} = 0.8$

	Idling		Fuel Economy Computer Program FB1213								
						Steady Speed mph					
	Speed rpm	Fuel Flow lbs/hr	City mpg	Suburban mpg	Customer Average mpg	30 mpg	40 mpg	50 mpg	60 mpg	70 mpg	30-70 mph Average mpg
Ford Production Engines											
302-2V 3-Speed Automatic	500	3.75	13.3	18.0	15.7	27.3	23.3	20.4	18.0	16.2	21.0
250-1V 3-Speed Automatic	600	2.68	12.9	19.6	16.3	27.0	25.4	22.9	20.1	17.6	22.6
Rankine Cycle Engines											
182-CID Clutch Drive, Single Speed	300	2.00	9.6	15.8	12.7	33.1	25.1	20.4	16.7	13.4	21.7
182-CID1.88 Ratio Speed-Up Gear	300	2.00	10.1	15.0	12.5	32.4	24.6	19.7	16.0	13.0	21.2
182-CID 2.77 Ratio Speed-Up Gear	300	2.00	10.3	15.2	12.8	32.3	24.8	19.5	15.7	12.9	21.0

I-1086

TABLE 5.2.4

VEHICLE PERFORMANCE PROJECTIONS

$IR_{max} = 0.29$

Full Torque Curve Used (No Allowance for Accessories)

Transmission	Gear Ratios	Engine Shift Speed	0 - 60 mph Acceleration Time Seconds	Gradability
Single Speed	1/1	-	17.1	19.1%
Two Speed	2.5/1 1/1	1800	12.5	56.3%
Two Speed	2.25/1 1/1	1800	12.4	49.1%
Two Speed	2.0/1 1/1	1800	12.5	42.4%
Two Speed	1.75/1 1/1	1800	12.8	36.1%
Two Speed	1.50/1 1/1	1800	13.2	30.2%

TABLE 5.2.5

VEHICLE PERFORMANCE PROJECTIONS

$IR_{max} = 0.29$

54 ft # Subtracted from Full Torque Curve at All Speeds for Accessories

Transmission	Gear Ratio	Engine Shift Speed	0 - 60 mph Acceleration Time Seconds	Gradability
Single Speed	1/1	-	19.6	17.0%
Two Speed	2.5/1 1/1	1800	14.4	48.9%
Two Speed	2.5/1 1/1	1800	14.5	42.5%
Two Speed	2.00/1 1/1	1800	14.6	37.2%
Two Speed	1.75/1 1/1	1800	14.9	31.9%
Two Speed	1.50/1 1/1	1800	15.5	26.7%

5.3 PACKAGING OF SYSTEM AND SYSTEM WEIGHT

For the present study, the criterion used in packaging the system has been that the entire system should be packaged in the engine compartment of the 1969 Ford Fairlane with only minor sheet metal and frame changes required for mounting of the Rankine-cycle components and with rear wheel drive. This approach was followed for the following reasons:

a. It permits use of the current production automobile chassis, with only minor changes, for constructing the first prototype.

b. The optimum system appears to be one in which the components are packaged in close proximity to each other, eliminating long fluid or vapor lines between separated components and permitting the engine to directly drive high power accessories such as the condenser fans and feedpump. Use of a regenerator increases the number of fluid and vapor runs required when components are separated.

While it appears desirable to package the complete Rankine-cycle system in one location in the vehicle, alternatives do exist for incorporating the system into an automobile which have not been investigated in this study. As an example, the fact that the Rankine-cycle system is composed of several components, each of which is relatively small compared to the equivalent internal combustion engine-transmission system, would make it particularly attractive to incorporate and package a front-wheel drive system for an automobile.

In Figure 5.3.1 the packaging of the major components for the 100 hp Rankine-cycle power plant in a 1969 Ford Fairlane engine compartment is illustrated. The engine-transmission assembly is installed in exactly the same relative position as the current IC engines in production in rear-wheel-drive automobiles. The much shorter engine provides room for the boiler and condenser to be placed in front of the engine as shown. The condenser is placed in the front of the car in the same relative position as the radiator of an IC car, facilitating movement of air through the condenser. The condenser is wider than the radiator for an IC engine, however, and some modification of the fender skirts will be required for incorporation of the condenser. The regenerator is positioned directly above the engine, permitting a direct run of the engine exhaust vapor into the regenerator. The regenerator is mounted directly to the engine by means of flanges cast into the engine block on the exhaust vapor lines and also serves as the oil separator for the system.

In the photographs of Figure 5.3.2, a complete mockup of the 100 horsepower system installed in tne engine compartment of a 1969 Ford Fairlane chassis is illustrated. The major components, as described in the component designs of Section 4, fit without difficulty, with room remaining for convenient placement of the power system and vehicle accessory components (including air conditioning). All power-driven accessories are included at the front with the condenser fans so that they may be driven by a single-power driveshaft. The accessory power drive must come from the rear of the engine, since only a single rotary shaft seal is desired. In the mockup, a flexible shaft is used for this purpose; an alternative is to use a solid shaft drive with universal joints.

The 1969 Ford Fairlane used a coil spring front suspension which restricts space in the engine compartment. Additional room, if required at a later date, could be obtained by use of a torsion bar suspension. However, for the 100 hp system, this modification is not necessary.

In Table 5.3.1, a tabulation of the total system weight is presented and compared with that of the 302-2V internal combustion system with 3-speed transmission.

The Rankine-cycle system is estimated to weigh about 150 lbs more than the internal combustion engine system. This additional weight may require use of a heavy duty suspension system in the front for optimum vehicle riding quality.

In Table 5.3.2, the system thiophene inventory is given. The total system inventory is 31 lbs. The lubricant inventory is estimated to be 4 quarts.

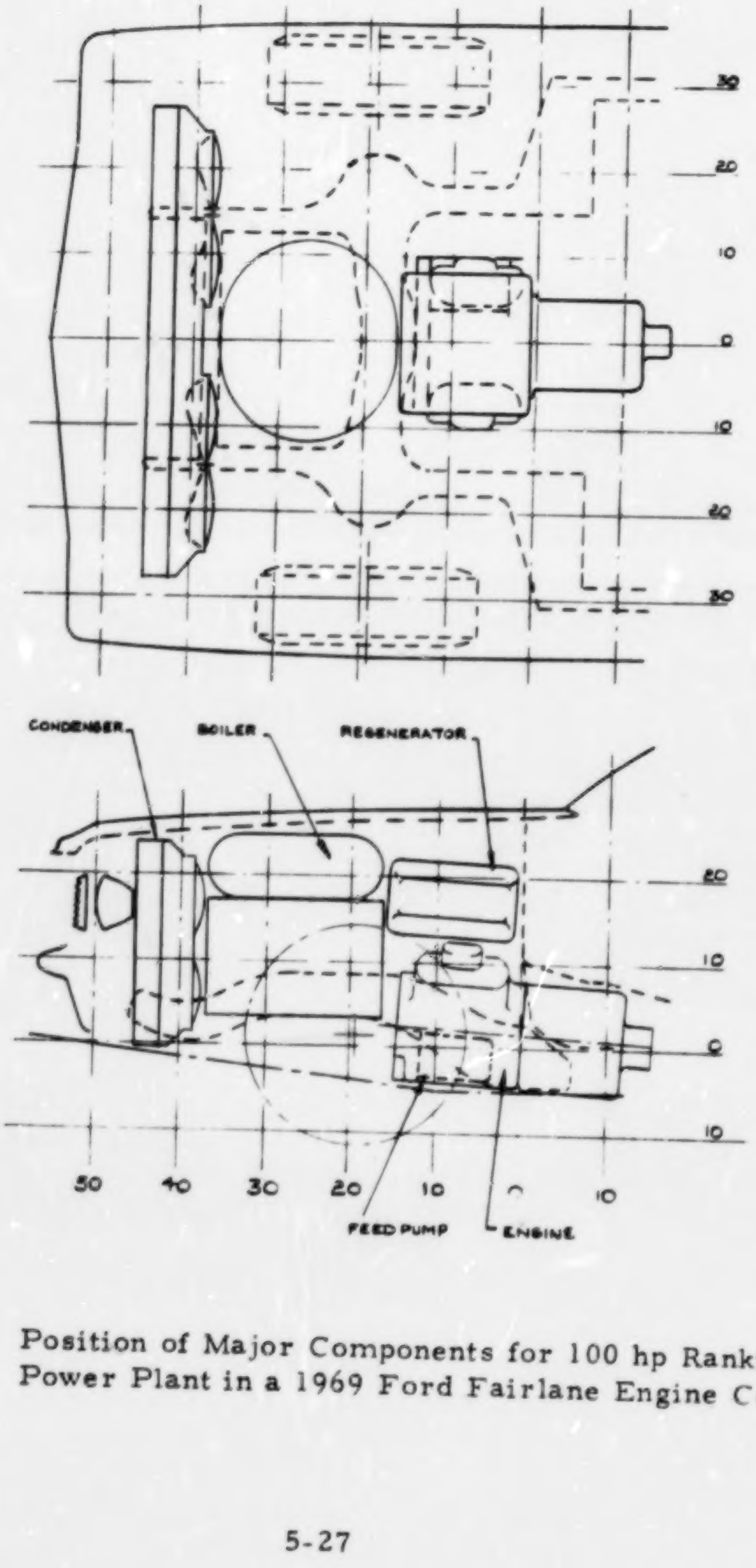
30
20
10
0
10
20
30
CONDENSER
BOILER
REGENERATOR
20
10
0
10
50
40
30
20
10
0
10
FEED PUMP
ENGINE

Figure 5. 3. 1 Position of Major Components for 100 hp Rankine-Cycle Power Plant in a 1969 Ford Fairlane Engine Compartment.

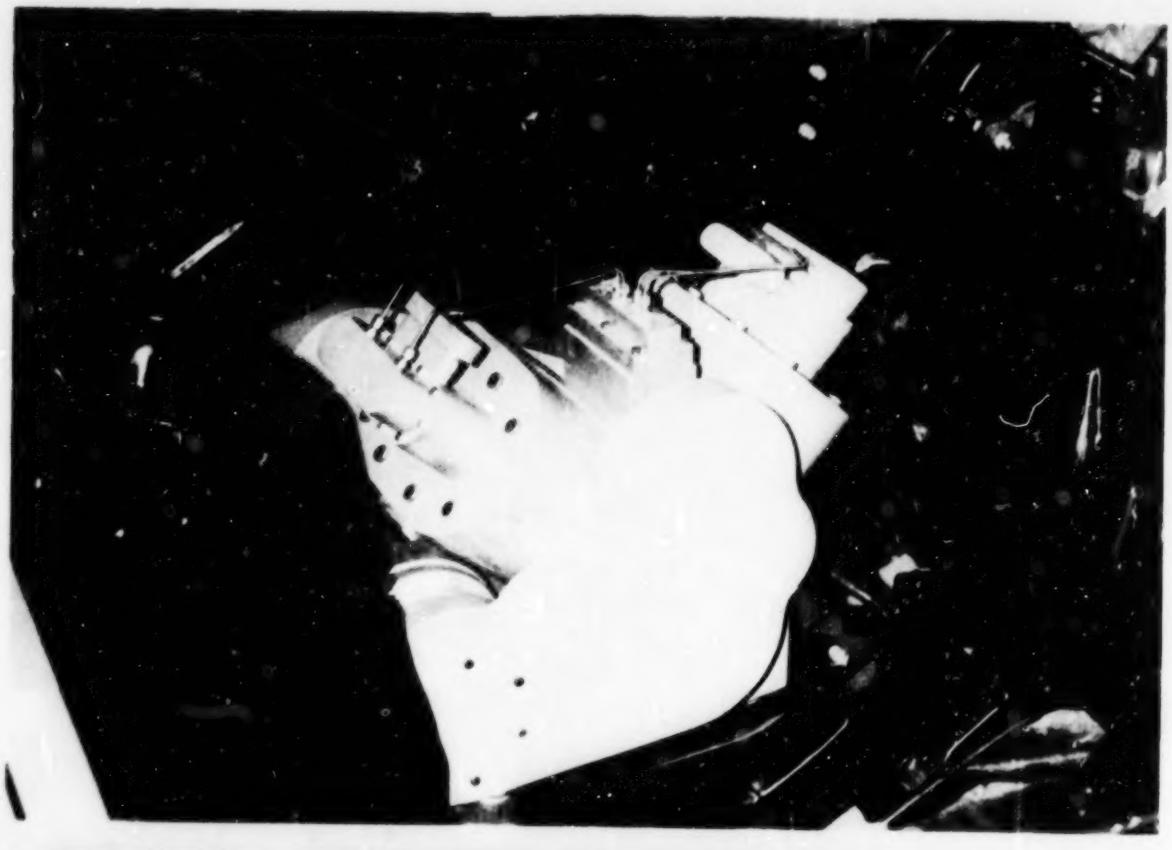

Figure 5.3.2 Photograph of Full-Size Mockup of 100 hp Rankine-Cycle Power Plant Installed in 1969 Ford Fairlane Engine Compartment.

a. Installation of V-4 Engine-Transmission Assembly Before Addition of Other Components.

b. Complete System Installed Including All Accessories.

TABLE 5.3.1

TABULATION OF TOTAL SYSTEM WEIGHT AND COMPARISON WITH 302-2V INTERNAL COMBUSTION SYSTEM WITH 3-SPEED TRANSMISSION

	Rankine Reference Design	302 Cu. In. V-8 with 3-Speed Automatic
Engine Expander Assembly	220	
Feedpump	45	
Engine Subsystem	265 lbs	479 lbs
Transmission	135 lbs	159 lbs
Burner-Boiler	273 lbs	
Regenerator	54 lbs	
Condenser	115 lbs	
Radiator with fan, connectors, and water		54 lbs
Controls, Exhaust, Electrical System, Accessory Drives, and other Miscellaneous Components	75 lbs	114 lbs
Working Fluid and Lubricant	40 lbs	
Total	957 lbs	806 lbs

TABLE 5.3.2

THIOPHENE INVENTORY IN SYSTEM

Boiler	11.2 lbs
Regenerator	10.6
Condenser	7.2
Lines	1.7
Engine and Feedpump	0.1
Total	30.8 lbs

5.4 EMISSION LEVEL FROM THE SYSTEM

Of great importance are the emission levels in gms/mile which can be expected from a Rankine-cycle propulsion system. The emission levels obtained with the burner developed at TECO for a 5 hp Rankine-cycle system have been used in making these projections. The control system for the burner will be designed to maintain optimum fuel/air ratio at any burning rate to minimize pollutant emissions. It is believed that the emission levels obtained from the TECO burner can be attained, with development, in the full-size burner for the automotive propulsion system.

In Table 5.4.1, projections are made of the emission levels in grams/mile from the Rankine-cycle propulsion system, using the measured emission data reported in Section 4.5. The projections are based on a fuel economy of 10.0 mpg. The projected customer average fuel economy for the Rankine-cycle system is 12.7 mpg, leaving a 2.7 mpg difference to account for driving accessories such as air conditioning, power steering, and power brakes, as well as a safety factor to account for any uncertainties in projected system fuel economy. The average emission levels used in the emission projections are:

Excess Air	33%
Unburned Hydrocarbons	15 ppm
Carbon Monoxide	60 ppm
Nitrogen Oxide	40 ppm

It is apparent that the Rankine-cycle system has the potential for exceeding the projected 1980 Federal Standards in gms/mile, as given in Table 5.4.1, being a factor of 5 lower in UHC, a factor of 13.4 lower in CO, and a factor of 1.6 lower in NO.

TABLE 5.4.1

PROJECTION OF MEASURED EMISSION DATA TO TECO RANKINE CYCLE PROPULSION SYSTEM

Excess Air = 33%
mpg = 10

Pollutant	Projected Emission Levels for TECO Rankine Cycle Propulsion System		Projected Federal Standards gms/mile		Uncontrolled IC Engine (1967)	IIEC Goals for IC Engine
	ppm, Exhaust Gas	gms/mile	1975	1980		
UHC	15	0.050	0.5	0.25	11.5	0.82
CO	60	0.35	11.0	4.70	85	7.1
NO	40	0.25	0.9	0.40	4	0.68

5.5 RELATIVE COST COMPARISON WITH 302-2V FORD ENGINE

Work on this task has been initiated with the Ford Motor Company, but is not available for this report. The cost information will be available in a later supplement to this report.

5.6 GENERALIZED COMPUTER MODEL

A generalized computer model for steady-state performance calculations has been completed, providing refinement of the performance calculations presented earlier. Any performance changes from use of this more detailed model will be second-order effects and should not affect the conclusions with respect to system performance. Typical printouts are illustrated in Tables 5.6.1, 5.6.2, and 5.6.3. This program uses detailed models describing the performance of all major components including engine, boiler, feedpump, regenerator, condenser, and condenser fans; pressure drops in connecting lines are also calculated. In the calculation, condenser pressure is not permitted to fall below approximately 10 psia, and the position in the condenser where 100% condensation is reached is controlled at a constant point in the condenser to insure constant fluid inventory in the system.

In Tables 5.6.1, 5.6.2, and 5.6.3, three runs from this program are presented. Comparison of the efficiencies from these tables with those given in Figure 5.1.2 for the same horsepower and engine rpm, as calculated with the preliminary performance program, indicates a slight increase in overall system efficiency for the low power (part load) operation and a decrease in high power (wide-open throttle) operation. For low power operation, the condenser is oversized, and thus operates at the minimum pressure (10 psia) allowed. This factor, coupled with an increased boiler efficiency, accounts for the improved efficiency under part-load conditions relative to the earlier calculations. At the higher power levels, the condenser pressure must be considerably higher for the required heat rejection so that the overall system efficiency is decreased slightly relative to the earlier calculations.

The results of Table 5.6.2 represent one of the poorer situations for condenser operation: full power output (IR = 0.29) at low vehicle speed (5.70 mph) with low engine speed (300 rpm). The condenser size used in this report is adequate for this condition with a condenser pressure of 62.8 psia, well below the design pressure of 100 psia. Thus, it can be concluded that the condenser is adequate for total condensation in the completely sealed system for any condition which can be encountered in operation of an automobile.

TABLE 5.6.1

PERFORMANCE PREDICTION FROM GENERALIZED MODEL

```
                  T      P      W           DT       DP       DW            Q        HP

14  BOIL IN    353.94 513.13 -60.36  BOIL 196.055   13.125 183.757       .551287
 1       OUT   550.00 500.00 123.40  LINE    .142    1.002    .000

 2  ENG  IN    549.86 499.00 123.40  ENG  152.415  436.035  32.699       .026508   28.13
 3       OUT   397.44  62.96  90.70  LINE    .013     .085    .000

 4  REGV IN    397.43  62.88  90.70  REGV  97.210     .126  29.971       .089916
 5       OUT   300.22  62.75  60.73  LINE    .006     .020    .000

 6  COND IN    300.21  62.73  60.73  COND  18.161    -.042 152.815       .458459
 9       OUT   282.05  62.77 -92.08  LINE    .000     .573    .000

10  PUMP IN    282.05  62.20 -92.08  PUMP   2.849  423.177   1.757                  2.07
11       OUT   284.90 514.62 -90.33  LINE   -.001     .288    .000

12  REGL IN    284.90 514.33 -90.33  REGL  69.041     .812  29.971       .089916
13       OUT   353.94 513.52 -60.36  LINE   -.000     .397    .000

CYCLE         FLOWRATE= 3000 LB/HR    KPH=  5.70             HP=  24.68          EFF=  9.68

ENGINE        PI= 496.85              PR= 159.67             V2=  .188           V3= 1.664

              DWIV=   .037            DWEV=   .000           Q1=  8.836          Q2=  4.380

              EFFTH= .7290            EFFME= .9574           EFFALL= .6979

              WFS= 32.736             WSHAFT= 22.845         WNET= 21.088

              INTAKE RATIO=.2900      RPM= 300

REGENERATOR   EFF= 85.78

FAN           AIR FLOW= 10193         FAN POWER=   .18       FAN DP=   .16       VEL DP=   .01

              FAN RPM=  600.0

CONDENSER     AIR IN= 95.00F          AIR OUT=263.16F        CORE DP=   .17IN    CORE POWER=   .08HP

              VAPOR=  2.0             COND= 52.5             LIGLID= 45.5        PSUB=  .96 PSI
                                                             ************

PUMP          EFFV= .9925             EFFME= .7850           EFFALL= .7792       PUMP WORK=  1.757

BOILER        HEAT IN=  .649 MBTU/HR  EFF= .8497
```

TABLE 5.6.2

PERFORMANCE PREDICTION FROM GENERALIZED MODEL

```
              T      P      H            DT      DP      DH          Q        HP
14  BOIL IN   254.75 511.59-102.59  BOIL 295.253  11.595 225.994
 1       OUT  550.00 500.00 123.40  LINE    .112    .789    .000    .598956

 2  ENG  IN   549.89 499.21 123.40  ENG  243.853 484.958  58.232
 3       OUT  306.03  14.25  65.17  LINE    .049    .272    .000    .012635  55.68

 4  REGV IN   305.99  13.98  65.17  REGV 109.539    .378  29.882
 5       OUT  196.45  13.60  35.29  LINE    .011    .051    .000    .079196

 6  COND IN   196.44  13.55  35.29  COND  22.786   -.806 169.535
 9       OUT  173.65  14.36-134.25  LINE    .000    .440    .000    .449323

10  PUMP IN   173.65  13.92-134.25  PUMP   2.879 473.338   1.774
11       OUT  176.53 512.74-132.48  LINE   -.001    .213    .000             1.85

12  REGL IN   176.53 512.53-132.48  REGL  78.217    .637  29.882
13       OUT  254.75 511.89-102.59  LINE   -.001    .298    .000    .079196
                                                                          = 9.68
CYCLF        FLOWRATE= 2650 LB/HR   MPH=  60.00          HP=  49.02          EFF= 17.73

ENGINE       PI= 441.33             PR=  34.18           V2=  .188           V3= 6.788
             DWIV=  1.005           DWEV=   .000         Q1=  4.767          Q2=  2.280
             EFFTH= .9026           EFFME= .9135         EFFALL= .8245
             WES= 59.238            WSHAFT= 48.841       WNET= 47.067
             INTAKE RATIO=.0750     RPM=1200

REGENERATOR  EFF= 87.16

FAN          AIR FLO= 40334         FAN POWER=   .00     FAN DP=   .00       VEL DP=  1.60
             FAN RPM=  400.0

CONDENSER    AIR IN=116.00F         AIR OUT=158.19F      CORE DP=  1.60IN    CORE POWER=  2.56HP
             VAPOR=  3.0            COND= 81.8           LIQUID= 15.2        PSUB= 2.05 PSI

PUMP         EFFV= .9700            EFFME= .7997         EFFALL= .7757       PUMP WORK=  1.776

BOILER       HEAT IN=  .703 MBTU/HR EFF= .8515

PROGRAM      FL= .2661              RES = .2003E-04      RES2=-.2193E-04     ITL=  9
```

NOT REPRODUCIBLE

TABLE 5.6.3

PERFORMANCE PREDICTION FROM GENERALIZED MODEL

```
                   T      P      H           DT       DP      DH           Q        HP
14  BOIL IN   345.02 530.18 -64.37  BOIL  204.976   30.178 187.772
 1       OUT  550.00 500.00 123.40  LINE     .728    5.128    .000    1.295203

 2  ENG  IN   549.27 494.87 123.40  ENG   155.969  433.811  33.927
 3       OUT  393.30  61.06  89.47  LINE     .067     .455    .000     .047988  73.10

 4  REGV IN   393.24  60.61  89.47  REGV   99.503     .543  30.518
 5       OUT  293.73  60.06  58.96  LINE     .021     .102    .000     .210502

 6  COND IN   293.71  59.96  58.96  COND   22.605   -1.077 155.533
 9       OUT  271.11  61.04 -96.58  LINE     .000    2.710    .000    1.072826

10  PUMP IN   271.11  58.33 -96.58  PUMP    2.726  404.517   1.688
11       OUT  273.83 537.16 -94.89  LINE    -.004    1.431    .000               4.57

12  REGL IN   273.84 535.72 -94.89  REGL   71.186    3.621  30.518
13       OUT  345.02 532.10 -64.37  LINE    -.003    1.926    .000     .210502

CYCLE          FLOWRATE= 6897 LB/HR   HPH=  15.21          HP=  61.60        EFF= 10.04

ENGINE         PI= 479.50             FR= 138.77           V2=  .190         V3= 1.705
               DRIVE=  .270           CHEV=  .000          Q1=  6.957        Q2=  3.380
               EFFTH= .7887           EFFME= .9504         EFFALL= .7496
               WFS= 34.197            WSHAFT= 25.633       WNET= 23.945
               INTAKE RATIO=.2400     RPM= 800

REGENERATOR    EFF= 88.30

FAN            AIR FLOW= 30568        FAN POWER=  3.30     FAN DP=   .97     VEL DP=   .11
               FAN RPM= 1600.0

CONDENSER      AIR IN= 95.00F         AIR OUT=232.08F      CORE DP=  1.07IN  CORE POWER=  1.36HP
               VAPOR=  2.0            COND= 84.8           LIQUID= 13.1      PSUB= 7.52 PSI

PUMP           EFFV= .9800            EFFME= .7751         EFFALL= .7596     PUMP WORK=  1.688

BOILER         HEAT IN= 1.561 MBTU/HR EFF= .8298
```

6. CONCLUSIONS

The major emphasis in developing the conceptual design in this study has been on the development of a system which has cost, performance, and convenience competitive with current automotive internal combustion engines and which uses the current state-of-technology to the fullest possible extent. The study has indicated that a Rankine-cycle automotive propulsion system with reciprocating expander and thiophene (or similar) working fluid has a very strong potential of meeting these objectives, with emission levels for all three of the major pollutants significantly less than the projected 1980 federal standards.

Specific conclusions arrived at in the conceptual design study are:

a. Packaging

A Rankine-cycle propulsion system competitive in performance with a 302 cubic inch displacement internal combustion engine can be completely packaged in the engine compartment of a 1969 Ford Fairlane with only minor internal sheet metal and frame modifications required.

b. Weight

The Rankine-cycle propulsion system has a total weight, as designed, approximately 20% greater than the equivalent internal combustion system. Design refinement of the Rankine-cycle system, coupled with changes required in the internal combustion engine system to meet future pollution requirements, will decrease this difference.

c. Engine Valving

A strong incentive exists for use of variable intake engine valving. A maximum intake ratio of 0.29 is preferable to 0.8 because of a reduction in the feedpump size and a reduction in the condenser cooling air requirement at low vehicle speeds with a relatively small effect on system performance.

d. Transmission

Use of a transmission which permits the main propulsion engine to idle at zero vehicle speed, allowing accessories to be driven directly by the engine, is preferable to use of a direct coupled engine requiring an auxiliary engine to drive the accessories. A two-speed transmission provides a significant decrease in the 0 - 60 mph acceleration time and an improved gradability of the vehicle relative to a single-speed transmission.

e. Condenser

A condenser-fan combination with sufficient capacity to handle the peak-load condensing requirements in the completely-sealed system is feasible for the 100 shp system using state-of-the-art technology.

f. Performance

A Rankine-cycle system of 100 shp net output (184 cubic inch engine displacement) with two-speed transmission is almost equivalent in acceleration performance and top speed to a 302 cubic inch displacement internal combustion system with three-speed transmission (advertised power of 220 hp).

g. Fuel Economy

With a maximum cycle temperature of 550°F, the customer-average fuel economy is 12.7 mph with the Rankine-cycle system and 15.7 mph with the 302 cubic inch displacement internal combustion engine system. A modest increase in maximum cycle temperature to 600°F for the Rankine-cycle system, coupled with reduced fuel economy anticipated for the internal combustion engine as tighter pollution emission restrictions are imposed, will decrease this difference.

h. Emission Level

Projections of current emissions data from burners suitable for use in compact Rankine-cycle systems indicate a significant reduction in pollutant emission levels below the current projected 1980 federal standards. Assuming, to be conservative, 10 mpg fuel economy, the UHC gms/mile is a factor of 5 below the projected 1980 federal standards, the CO gms/mile a factor of 13.4 below, and the NO gms/mile a factor of 1.6 below.

i. Cost

A detailed, large-volume, manufacturing cost estimate for the system is being prepared by the Ford Motor Company for the conceptual design and will be available at a later date. The cost estimate will be reported as a ratio relative to the Ford 302 cubic inch internal combustion engine system with three speed automatic transmission.

j. Reliability and Maintenance

Since only limited test experience is available at this date, no quantitative information on these factors is available. However, the approach followed in development of the system is similar to that of hermetically sealed air-conditioning systems. It is expected that, as in the air conditioning systems, the system failures and maintenance requirements will occur primarily from control, motor, and accessory failures rather than from failure of the major mechanical components such as the engine or feedpump.

k. Working Fluid

The primary consideration in the conceptual design and in the first experimental prototype is demonstration of a low cost Rankine-cycle system competitive with the internal combustion system in performance and drivability, using the current state-of-the-art to the fullest extent. Thiophene, in our judgement, best fulfills these goals. In view of the flammability and toxicity of thiophene, a question does exist regarding its suitability for large scale use in automobiles for the general public. It is, therefore, strongly recommended that a comprehensive search and development effort be initiated for a fire resistant and low toxic working fluid which retains the otherwise desirable characteristics of thiophene.

APPENDIX A

PARAMETERS FOR CHARACTERIZING FLAMMABILITY CHARACTERISTICS OF MATERIALS

Flash Point

The lowest temperature at which a liquid will give off flammable vapor at or near its surface. This vapor forms an intimate mixture with air, and it is this mixture which ignites. The flash point of liquids is usually determined by the Standard Method of Test for Flash Point with the Tag Closed Cup Tester (ASTM D56-52). The Interstate Commerce Commission uses the Tag Open Cup Tester, giving results 5-10°F higher (less flammable).

Fire Point

The lowest temperature at which a mixture of air and vapor continues to burn in an open container when ignited.

Autoignition Temperature

The lowest temperature at which a material will self-ignite and sustain combustion in the absence of a spark or flame.

Explosive Range or Flammability Limits

Range of concentration of material vapor in air, expressed as per cent by volume, over which the vapor-air mixture will burn when ignited. The values are generally given for normal conditions of temperature and pressure.

Hazard Rating of Flammable Fluids

(1) Interstate Commerce Commission

Flash Point $\leq$ 80°F	High Fire Hazard
Flash Point 80-350°F	Moderate Fire Hazard
Flash Point >350°F	Slight Fire Hazard

(2) National Fire Protection Association

Flash Point ≤ 20°F	High Fire Hazard
Flash Point 20-70°F	Moderate Fire Hazard
Flash Point 70-200°F	Slight Fire Hazard
Flash Point >200°F	Not generally called flammable

APPENDIX B

API TOXICOLOGICAL REVIEWS

THIOPHENE AND DERIVATIVES
AND
GASOLINE

I-1105

API TOXICOLOGICAL REVIEW

THIOPHENE AND DERIVATIVES

SEPTEMBER 1948

W.G.K.
APR 11 1958

Note: This review summarizes the best available information on the properties, characteristics, and toxicology of *thiophene and derivatives.* It offers suggestions and tentative recommendations pertaining to medical treatments, medical examinations, and precautionary measures for workers who are exposed to thiophene and derivatives. It was prepared at the Harvard School of Public Health, Boston, Mass., under the direction of Professor Philip Drinker. The review has been accepted for publication by the Medical Advisory Committee of the American Petroleum Institute. Anyone desiring to submit additional information or proposed changes for consideration prior to re-issuance of this review is requested to send them to the American Petroleum Institute.

This review was prepared by Marshall Clinton, M. D.

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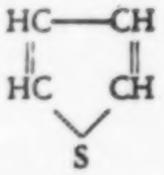
B-1

Price 25 Cents

TOXICOLOGICAL REVIEW OF THIOPHENE AND DERIVATIVES[*]

I. Substance

Thiophene.
Formula: C_4H_4S.
Structural formula:



Molecular weight=84.13.
Synonym: thiofuran.

II. Properties and Characteristics[1, 2, 3a]

Melting point =minus 38 deg C (−36.4 deg F).
Boiling point =84 deg C (183.2 deg F).
Refractive index=1.5285.
1 mg per liter =291 ppm; 100 ppm=0.344 mg per liter.

Thiophene is a clear colorless heterocyclic compound encountered as an important contaminant of benzene. It is insoluble in water, but is readily soluble in alcohol, ether, benzene, and most hydrocarbons. Thiophene is difficult to separate from benzene by physical means because of their similar boiling points, but can be separated fairly readily after reaction of the more reactive thiophene with other substances, such as mercury. Thiophene can be obtained from crude benzene, and now can be synthesized without excessive difficulty. Thiophene is highly reactive, and is readily nitrated, sulfonated, halogenated, or mercurated. It can be made to undergo ketone formation or aminomethylation without difficulty. It is usually removed from benzene by sulfuric-acid treatment.

III. Probable Sources of Contact

Contact with thiophene may occur due to leaks occurring in the course of its handling or manufacture. Contact may result from handling crude coal-tar benzene, as this contains up to 0.5 per cent thiophene. It is not possible to state the most probable sources of exposure, as this substance is, commercially, relatively new, and its uses are ill-defined but growing.

IV. Toxicology

a. General Considerations

A considerable amount of study has been devoted to investigations of the toxicology and pharmacology of thiophene and its derivatives. Most of the latter studies have been comparisons of the thio homologues of organic substances of known and, usually, fairly marked pharmacological activity. Thiophene in fairly high concentrations has, according to most authors,[4, 5] an acute narcotic effect greater than equal concentrations of benzene. Flury and Zernik report that the inhalation by mice of 2,900 ppm of thiophene results in loss of consciousness and, in some instances, death; whereas similar concentrations of benzene can be tolerated without difficulty. Concentrations of 8,700 ppm of thiophene caused death of mice in 20 min to 80 min; whereas benzene produced no such effect.

The acute toxic action of thiophene appears to be exerted primarily on the central nervous system. It has a selective action on the equilibrium centers of the cerebrum and cerebellum, producing severe ataxia following repeated injections.[5, 6] Thiophene produces fairly diffuse changes in the cerebellum, particularly the vermis, characterized by degeneration of nerve cells in these areas. There may be superimposed vascular changes.[7, 8] The metabolism of thiophene is poorly understood, although it is stated that 5 to 12 per cent is recovered in the urine in conjugated form. However, there is no increase in the conjugated sulfates, and the total sulfate excretion decreases after administration of thiophene.[9]

b. Acute Effects

As already noted, acute exposure to high concentrations of thiophene results in nervous-system depression. Repeated daily injection of 2 g of thiophene in dogs results in locomotor ataxia and paralysis,[5] rather similar to those noted in severe CS_2 poisoning. The effects of thiophene on humans has not been described.

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[*] Prepared under the auspices of the Subcommittee for Permissible Concentrations of Toxic Substances in the Petroleum Industry.

[a] Figures refer to bibliography on p. 3.

c. Chronic Effects

No reports on the chronic effects of repeated or prolonged exposure to low concentrations of thiophene are available.

d. Safe Limits

Extreme concentrations of thiophene are obviously intolerable, as they produce acute poisoning. No information is available, however, on the effects of repeated exposure to lesser concentrations, such as 100 to 1,000 ppm. Therefore, no safe limits have been or can be promulgated at present.

V. Treatment

No information on the possible therapy of thiophene poisoning is available.

VI. Examinations

The present state of knowledge concerning thiophene does not permit the establishment of any special pre-employment or periodic examinations. It appears sensible to employ only men in good health to work with thiophene and to re-examine them frequently for possible evidences of blood dyscrasia or neurologic disturbances, but these measures may prove unnecessary.

VII. Precautionary Measures

Thiophene should be handled with extreme care, in closed systems or with adequate ventilation, until its chronic toxicity is established or shown to be absent.

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API TOXICOLOGICAL REVIEW

GASOLINE

FIRST EDITION, 1967

The information and recommendations contained in this publication have been compiled from sources believed to be reliable and to represent the best current opinion on the subject. No warranty, guarantee, or representation is made by the American Petroleum Institute as to the absolute correctness or sufficiency of any representation contained in this and other Toxicological Reviews, and the Institute assumes no responsibility in connection therewith; nor can it be assumed that all acceptable safety measures are contained in this and other Toxicological Reviews, or that other or additional measures may not be required under particular or exceptional conditions or circumstances. The American Petroleum Institute, as sponsor of this review, takes no position as to whether or not any method contained herein is covered by an existing patent, nor as to the validity of any patent alleged to cover any such method. Furthermore, nothing contained in this review grants any right, by implication or otherwise, for the manufacture, sale, or use in connection with any method, apparatus, or product covered by letters patent.

This review was prepared by the Committee on Toxicology and accepted by the Central Committee on Medicine and Health. Anyone desiring to submit additional information or proposed changes for consideration prior to reissuance of this review is requested to send them to the American Petroleum Institute.

AMERICAN PETROLEUM INSTITUTE

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API TOXICOLOGICAL REVIEW OF GASOLINE

Substance

Gasoline is a refined petroleum product suitable for the operation of an internal-combustion engine. It is a complex mixture of hydrocarbons to which are usually added antiknock agents, inhibitors, and dyes. The hydrocarbons present are primarily paraffins, naphthenes, aromatics, and olefins. Widely varying amounts of the individual hydrocarbons are contained in typical gasoline blends, depending on such factors as the origin of the blending streams, seasonal requirements, and intended use.

II. Properties and Characteristics

Distillation range (by ASTM D86) = 32 C to 225 C (90 F to 437 F)

Specific gravity = approximately 0.71 to 0.77

Flash point (Tag closed cup) = May be as low as −45 C (−50 F)

Explosive limits = 1.3 to 6.0 percent by volume in air

Because the composition of gasoline is variable, it is possible to make only general statements on the properties and characteristics.

It is an extremely flammable liquid and is water-white to straw-tint in color before the addition of dyes.

III. Uses and Probable Sources of Contact

Gasoline is intended for use as a motor fuel. Although sometimes used as a cleaning agent and as a substitute for other solvents, it should not be used other than as a motor fuel unless adequate precautions are taken to control both the potential health hazard and the fire hazard.

In refineries, exposure to gasoline vapors may occur at process units, in the repairing of refinery equipment, in the cleaning of storage tanks, during the gaging of tanks, and in the laboratory. Potential exposures may also occur in the filling of tank cars, tank trucks, drums, storage tanks, and in the fueling of automobiles and aircraft.

Gasoline exposures also occur from the use of gasoline as a cleaning agent or as a solvent substitute or from careless handling and storage which may result in accidental ingestion.

IV. Toxicology

a. General Considerations

Gerarde[18] points out that although the hydrocarbon composition of gasoline has changed over the years, its basic pharmacology and toxicology have not altered significantly. The signs and symptoms of intoxication from acute exposure to gasoline are similar to those for an exposure to heptane, namely, marked vertigo, inability to walk a straight line, hilarity, and incoordination.

Depending on the methods of manufacture and on the blending components, gasoline may contain benzene. In cases of repeated exposure to significant amounts of gasoline vapor, the potential hazard of benzene exposure may have to be considered. Benzene is unique among the hydrocarbons in its ability to depress the hemopoietic system.[2]

Numerous additives may be present in the many branded gasolines. In general, these materials are added in very low concentrations and do not contribute significantly to the toxicity of gasoline by inhalation and skin contact. In the case of tetraethyllead (TEL), Kehoe[3, 4] states that the low concentration of TEL in gasoline effectively prevents the absorption of significant quantities of TEL through the skin. Similarly, the vaporization of TEL from gasoline is so low at ordinary temperatures as to preclude its presence in more than minute quantities in gasoline vapor. Hence, persons dispensing gasoline as a motor fuel have no significant exposure to TEL. However, if gasoline containing TEL is spilled, sprayed, or otherwise vaporized in unventilated or enclosed spaces, the concentration of TEL may exceed safe levels. Similarly, a significant exposure to TEL vapors occurs in the removal of sludge from storage tanks which have contained leaded gasoline.

Because tetramethyllead (TML) is more volatile than TEL, Kehoe, et al.,[5] recently investigated the handling of gasoline containing this material. In a study of both refinery workers and service station attendants, it was concluded that exposure of the various groups to TML, under the prevalent environmental conditions of the occupations, is negligible. The comments made in the preceding paragraph concerning the hazard of TEL gasolines in unventilated or confined spaces also apply to gasolines containing TML.

b. Acute Toxicity

Inhalation

Browning[6] states that many severe or fatal cases from inhalation of gasoline vapors are reported but these

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[*] Figures refer to BIBLIOGRAPHY on p. 5.

have almost always involved men who entered tanks containing high concentrations of gasoline vapors. According to von Oettingen,[7] inhalation of very high concentrations of vapors may cause sudden loss of consciousness, coma, and sudden death. Browning[6] and von Oettingen[7] report the following signs and symptoms: In severe cases, delirium with cyanosis, coma, tonic and clonic convulsions, shallow and stertorous respiration, and a thready pulse. Vomiting, inward strabismus, contracted pupils, and loss of reflexes have also been observed. In less severe cases, headache, flushing of the face, nausea, mental confusion and depression, anorexia, blurring speech, and difficulty in swallowing have been observed.

Gerarde[1] states that there are a number of reports in the clinical literature which indicate that very acute hydrocarbon intoxication may cause central nervous system sequelae, such as convulsions or seizures, several months after the initial acute exposure.

Wang and Irons[8] report a fatal case of gasoline intoxication in a man who entered an unpurged aircraft wing tank in which the concentration of vapor was estimated to be 0.5 to 1.6 percent. Approximately 5 min after entry, he was found unconscious within the tank and was quickly removed. Artificial respiration was applied immediately but the patient died en route to hospital. Autopsy revealed acute pulmonary edema, acute exudative tracheobronchitis, passive congestion of the liver and spleen, and early hemorrhagic pancreatitis. The authors believe that the clinical history and pathological findings were entirely compatible with a diagnosis of death due to hydrocarbon poisoning.

Necropsy, on a youth who died while using a ladle to fill a 2-gal can with gasoline from a supply barrel, revealed nothing abnormal other than raw areas of skin on the wrists and upper arms. Concentration of vapor was estimated to be 500 ppm to 30,000 ppm.[9]

MacLean[10] reports three cases of fatal aplastic anemia which are assumed to have occurred after siphoning gasoline containing benzene or inhaling its vapors. As the benzene content of the gasoline was approximately 10 percent, benzene rather than the gasoline itself is thought to be the cause of the anemia.

Ainsworth[11] reports the following postmortem findings in a young boy found unconscious in a pool of gasoline: Hyperemia was present in all organs examined; lungs showed considerable edema, some intraalveolar hemorrhage, and necrosis of alveolar walls; superficial epidermis was loose and could be stripped off with ease.

Ingestion

The acute oral toxicity of gasoline for rats ranges from 10 to 35 g per kg of body weight.[12] Although serious poisoning of humans may result from the ingestion of 20 g to 30 g of gasoline, the usual fatal dose for adults is approximately 350 g. With children 10 g to 15 g may be fatal.[13] The variation in susceptibility is caused by a number of factors, including the presence of food in the stomach and, most importantly, whether respiratory aspiration occurs.

According to von Oettingen,[7] the ingestion of gasoline causes a burning sensation in the mouth, pharynx and chest, and intense irritation of the gastrointestinal tract, with vomiting, colic, and diarrhea. Dizziness, unconsciousness, and coma may also result. In nonfatal cases, bronchitis, pneumonia, and nephritis may develop.

In cases of ingestion, it is generally believed that the resulting pneumonitis, if present, is caused by aspiration of gasoline into the lungs. Gerarde[14] has shown that the aspiration of as little as 0.2 ml by rats causes instantaneous death.

c. Chronic Toxicity

Gerarde[1] states that in service station attendants and garage workers who are exposed repeatedly to low concentrations of gasoline vapors with brief exposures to higher concentrations, there is no conclusive evidence of harmful health effects due to exposure to gasoline vapors.

Browning[6] expresses the opinion that reports of chronic poisoning are few and vague; most of the authorities that mention them quote from earlier works rather than from personal experience. Machle[15] reports chronic poisoning to be rare. In his study, 2,300 refinery workers showed no symptoms, nor did service station attendants, tank wagon drivers, etc. However, he states that "barrel fillers," who were exposed to concentrations which might well be intolerable to many people, showed signs of malnutrition, pallor, anorexia, nausea, nervousness, and low hemoglobin in a high proportion of the individuals involved.

MacLean[10] reports that an oil company employee developed hemolytic anemia and myelofibrosis after 12 months' exposure to gasoline vapor resulting from spills. He also reports a case of thrombocytopenic purpura in a man who had cleaned metal parts in gasoline over a 2-year period. In the first case, the benzene content of the gasoline was less than 1 percent; in the second case, benzene content may have been as high as 10 percent.

Sterner[16] reports on a group of painters who were exposed to vapors released during spray painting with gasoline-diluted paints. The concentration of aromatic hydrocarbons in the vapor was from 300 ppm to 800 ppm, and it was assumed that the total hydrocarbon concentration would be five to ten times more than this range. The chief symptoms were headache, nausea, weakness, mental depression, anorexia, and inability for sustained attention and activity. One case showed a tremor and weakness of the arms and legs with multiple fibrillary twitchings on fatigue. A significant decrease was noted in hemoglobin, erythrocytes, and blood cell volume values, with an increase in mean corpuscular hemoglobin, mean corpuscular volume, and reticulocyte count.

Oldham [17] reports on a 17-year-old girl who indulged in the habit of repeated self-intoxication with gasoline vapor, three or four times a week, over a period of two years. The patient described the effects as consisting of "dreams" followed by giddiness, nausea, and vomiting. There were no apparent chronic effects.

Drinker, et al.,[18] state that exposure to 1,000 ppm for 1 hr caused slight dizziness, nausea, and headache in human subjects. When the concentration reached 2,600 ppm, all subjects were drunk and somewhat anesthetized. In a group exposed to 160 ppm and 270 ppm, for 8 hr with a midday break of 1 hr, the most distinctive symptoms were irritation of the eyes and throat.

Davis, et al.,[19] exposed human subjects to nonleaded gasoline of the following approximate compositions:

	Paraffins (Percent)	Naphthenes (Percent)	Aromatics (Percent)
Sample A	25	30	40
Sample B	40	35	20
Sample C	30	5	65

Eye irritation was the only effect observed after ½-hr exposure to vapor. Irritation increased as the concentration increased from 200 ppm to 500 ppm to 1,000 ppm. There was no significant difference in the irritating potential of the three gasolines.

Skin Effects

Gasoline is a primary skin irritant and prolonged contact may dry and defat the skin with resultant dermatitis. Sensitization [20] to gasoline has been reported but is not common.

Eye Effects

Gasoline causes smarting and pain on splash contact with the eyes, but only slight transient corneal epithelial disturbances.[21]

Exposure of rabbit eyes to both leaded and unleaded gasoline caused the conjunctiva to become moderately edematous and hyperemic. However, the injury was superficial and transient. There was no difference in reactions between the leaded and unleaded gasoline.

d. Permissible Limits of Exposure

The threshold limit value adopted by the American Conference of Governmental Industrial Hygienists for an 8-hr day is 500 ppm.[22]

V. Treatment

Inhalation

The first aid care of a victim of acute vapor exposure requires his immediate removal from the contaminated atmosphere. Rescuers should take suitable precautions to prevent their being overcome by high concentrations of vapors. If breathing is interrupted, artificial respiration should be applied immediately. A physician should be called. Medical treatment is symptomatic. No specific measures are advocated.

Ingestion

Ingestion of gasoline is rarely encountered in industry. In the home it may be accidentally swallowed and usually by very young children.

A person who has ingested gasoline should be given olive oil or some other vegetable oil orally to retard absorption of the gasoline. Gastric lavage and the induction of vomiting are not advisable because of the possibility of aspiration of gasoline and the subsequent development of chemical pneumonia. The use of oxygen and antibiotics prophylactically for the prevention of secondary bacterial pneumonia may be indicated.

VI. Examinations

Special health examinations for the determination of toxic exposure are unnecessary for most employees engaged in the manufacture, distribution, and sale of gasoline as a motor fuel.

However, persons who may be repeatedly exposed to significant amounts of gasoline vapors or mists should be given a preemployment examination, and those with evidence of blood dyscrasias or serious systemic disorders should be excluded from such work. Periodic reexaminations should be carried out at intervals to be determined by the physician in charge. Particular attention should be paid to evidence of eye irritation, dermatitis, or symptoms related to the nervous system.

VII. Precautionary Measures

In gasoline manufacture, there is very little actual exposure to gasoline vapor because the process is carried out in a closed system.

Before entering tanks, vessels, or other confined space, adequate ventilation should be provided to maintain the gasoline vapor concentration at a safe level. In the cleaning and repairing of storage tanks in leaded gasoline service, special precautions are also necessary to control the potential hazard from the lead antiknock compounds that may be present.

Adequate local exhaust ventilation should be provided at drum-filling operations conducted indoors.

Leaded gasoline should be used only as a motor fuel.

Gasoline should never be siphoned by mouth.

Gasoline should be stored in a cool, well-ventilated place. To avoid accidental ingestion, it should be stored in clearly marked containers, well out of the reach of children.

If unleaded gasoline is used as a solvent substitute, adequate ventilation should be provided, as well as the other necessary precautions to prevent fire and explosion.

Repeated or prolonged skin contact should be avoided. If such contact is necessary, protective clothing and gloves should be worn. Goggles may be worn as protection against accidental splashes.

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